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CELCAP: A Computer Model for Cogeneration System Analysis

G. C. Birur

July 1985

Prepared for
Naval Civil Engineering Laboratory
Port Hueneme, California
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16. Abstract

A description of the CELCAP cogeneration analysis program is presented. A detailed description of the methodology used by the Naval Civil Engineering Laboratory in developing the CELCAP code and the procedures for analyzing cogeneration systems for a given user are given. The four engines modeled in CELCAP are: gas turbine with exhaust heat boiler, diesel engine with waste heat boiler, single automatic-extraction steam turbine, and back-pressure steam turbine. Both the design point and part-load performances are taken into account in the engine models. The load model describes how the hourly electric and steam demand of the user is represented by 24 hourly profiles. The economic model describes how the annual and life-cycle operating costs that include the costs of fuel, purchased electricity, and operation and maintenance of engines and boilers are calculated.

The CELCAP code structure and principal functions of the code are described to show how the various components of the code are related to each other. Three examples of the application of the CELCAP code are given to illustrate the versatility of the code. The examples shown represent cases of system selection, system modification, and system optimization.

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FOREWORD

The work reported in this document is one part of the total effort to upgrade the Civil Engineering Laboratory Cogeneration Analysis Program (CELCAP) originally developed by the Naval Civil Engineering Laboratory (NCEL), Port Hueneme, California. This total effort to upgrade CELCAP consists of three steps: (1) to provide NCEL with the instructions on how to improve CELCAP and convert it into a "user-friendly" program, and to document three of the heat engine models used in CELCAP; (2) to upgrade the CELCAP code according to the recommendations made in Step 1 and develop a program description document; (3) to test the upgraded version of CELCAP and to develop program user documentation. The work presented in this report represents the work performed under Step 2.

The author thanks Dr. Richard Lee, the NCEL Project Manager, and Dr. Elliot Framan, the JPL Program Manager, for their support and encouragement during this study. Several people helped the author during the preparation of the report. Dr. John Roschke and Toshio Fujita reviewed a draft of the report and offered many helpful comments and suggestions. Marion Rice diligently typed the whole report including all the equations and flow charts. Charlotte Marsh edited the report. The author gratefully acknowledges the support given by all the above people.

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SECTION I

INTRODUCTION

Cogeneration energy systems can produce electricity and steam more efficiently and cheaply than conventional energy systems for many applications. This efficiency, combined with the recent increases in fuel costs, has made many industrial, commercial, and institutional users consider cogeneration systems as a viable on-site power plant. In the past, the lack of design and analysis tools for cogeneration systems limited potential users from exploring the feasibility of different cogeneration systems. The Naval Civil Engineering Laboratory (NCEL) realized the need for a tool for analyzing the feasibility of cogeneration systems for the Naval bases and developed a computer code for cogeneration analysis. This code, called Civil Engineering Laboratory Cogeneration Analysis Program (CELCAP), was developed between 1979 and 1981 and has been successfully used for examining cogeneration options for a number of military facilities. The program description of the CELCAP code is documented in this report.

A. BACKGROUND

The cogeneration systems are attractive to many industrial, commercial, and institutional users for several reasons. Chief among these is that a cogeneration system produces both electricity and steam simultaneously with a single energy source. This makes it cheaper to operate than a conventional system with steam produced in a boiler and electricity generated and/or purchased separately. Some of the other reasons are: (1) A cogeneration system on-site ensures a reliable supply of electricity and steam compared to buying it from a utility; (2) the tax and regulatory laws are favorable to cogeneration systems; (3) the ability of a cogeneration system to tie the on-site power plant to the utility is advantageous to the utility and the user; and (4) the inability of the local utility to expand and supply large power demands of a new industry can be solved by the cogeneration plants.

During the 1970s many Naval bases were interested in installing cogeneration systems. Some others, which already had cogeneration systems on their bases, were interested in expanding their capacities or modifying their existing systems. As a result, the NCEL was requested by the Naval Facility Engineering Command to examine Naval cogeneration energy systems. Some early work was done by Cooper (Reference 1). He outlined the different cogeneration options and presented an economic analysis methodology for comparing different cogeneration options. In 1981, Cooper presented a procedure for analyzing the performance of combustion turbine/exhaust boiler cogeneration systems (Reference 2). Based on these two pieces of work, Cooper and Lee (Reference 3) developed a computer program between 1979 and 1981 that was called CELCAP.

The CELCAP code was initially used by Cooper and Lee to analyze cogeneration energy systems for many Naval bases (References 3 through 6). In his evaluation of cogeneration options for the Naval hospital, Cooper (see Reference 3) used the CELCAP code to examine six cogeneration systems. These systems were single or multiple combinations of diesel with exhaust boilers or gas turbine with exhaust boilers. In his study on cogeneration options for a Naval base in Florida, Lee (see Reference 4) analyzed four different cogeneration concepts using the CELCAP code. These concepts consisted of back-pressure steam turbine combinations, diesel engines with waste heat boiler combinations, gas turbines with waste heat boilers combinations, and an automatic-extraction steam turbine combination. In their study on the impact of air-conditioning switch-over on the base energy system, Lee and Cooper (see Reference 5) used the CELCAP code to analyze the cogeneration energy systems at the Naval base at Pensacola, Florida. The change in electric and steam demand because of the air-conditioning switch-over is used in the CELCAP code along with engine information to analyze the cogeneration system performance. Lee, using the CELCAP code, examined the cogeneration potential at Marine Corps Development and Education Command at Quantico, Virginia, (References 6 and 7) and at Naval Construction Battalion Center, Port Hueneme, California (Reference 8).

The CELCAP code is operational and available to engineers for analyzing cogeneration energy systems. However, it has been used by only a few engineers to analyze cogeneration systems for some of the Naval bases. Moreover, engineers who are not associated with the Navy have not yet used it. This report is the program descriptive document of CELCAP. This report and the user's manual that will be developed later will provide a complete set of documents to assist engineers in the use of the CELCAP code. The user's manual to be developed will be an updated version of an earlier report (Reference 9). One of the purposes for writing this report is to provide more detailed documentation on CELCAP so that more engineers will be able to use the code in the future for their cogeneration application studies.

B. COGENERATION ANALYSIS COMPUTER PROGRAM

The CELCAP code was developed by NCEL to analyze cogeneration energy systems. While developing the code, NCEL was primarily concerned with the cogeneration systems that are commonly employed at Naval bases. These cogeneration systems are usually built around any one of the three types of prime movers - steam turbines, gas turbines with waste heat boiler, or diesel engines with waste heat boilers. The performance models of these engines are included in the code. Also included in the code are the economic models that compute the annual and life-cycle operating costs in the code. Given the electric and steam loads that have to be met by the cogeneration system, the CELCAP code computes the annual and life-cycle operating costs for meeting these loads.

The CELCAP code is organized in such a way that a wide variety of cogeneration system arrangements can be analyzed with ease. Information on the cogeneration system to be analyzed is provided to the code as an input data set. The input data contain information on the characteristics of each engine, the boiler, and the economic parameters for the energy system. Also included in

the input data are the hourly electric and steam loads that should be supplied by the cogeneration system. The input data are read by the program first. A flow diagram of the CELCAP code is shown in Figure 1-1. The first step in the code is reading of the input data on the number of engines in the system and how they should be operated. In the next step, the program calculates the limiting electric and steam generating capacities of the engines in the system. Once this is done, the performance of the cogeneration system is calculated on an hourly basis for meeting the electric and steam loads of the user. In the last step of the code, the annual and life-cycle operating costs of the system are evaluated using the economic parameters provided in the input data.

There are four types of cogeneration energy system models included in the CELCAP code. They are based on the following engines: gas turbines, diesel engines, automatic-extraction steam turbines, and back-pressure steam turbines. The gas turbine and diesel engine cogeneration systems have a waste heat recovery boiler that produces steam from the exhaust gases of the engines. This low-pressure steam, which is usually below 200 psig, is exported from the power plant to various steam users. In the case of automatic-extraction and back-pressure steam turbines, the partially extracted or expanded steam from the turbine, which is at low pressure, is exported out of the power plant. All four types of cogeneration systems modeled in the CELCAP code are topping cycle systems. A cogeneration system consisting of any combination of these four types (up to a maximum of five engines) can be evaluated by CELCAP. There are three control modes in which the engines can be operated. These control modes are: (1) constant operation at maximum allowable capacity, (2) modulation with the electrical load, and (3) modulation with the steam load.

To evaluate a cogeneration energy system using CELCAP, several types of information about the system have to be supplied to the code. Broadly speaking, the input data can be classified into three groups. The first group involves engine and boiler characteristics. The engine information includes the engine capacity and its part-load characteristics. The boiler data include the boiler efficiency and information on temperature, pressures, and enthalpies of the steam. The second group of data concerns the hourly electrical and steam loads that should be provided by the cogeneration system. The electric and steam hourly load profiles for a weekday and a weekend day are provided for each month of the year. The third group of data concerns the economic parameters needed to calculate the annual and life-cycle cost of operating the cogeneration system. For this, information on electric utility rate, fuel price, escalation rates, and discount rate is provided.

The output from CELCAP can be obtained either in a brief or detailed form. In the brief form, the output consists of the important input information on the engines, steam conditions, utility rates, and fuel prices, as well as a monthly summary of the on-site electricity and steam generation and the purchased electricity. Also provided in the brief printout are the life-cycle operation costs of fuel, operation and maintenance, and electric power. The detailed printout includes all the information of the brief printout, plus a great deal more. The hourly operating capacity of the engines and boilers is printed for 2 days in each month of the year. The maximum hourly total output and fuel consumption for each month are also printed. The hourly electric demand and supply are plotted, along with the hourly steam demand and supply for each month of the year.

C. CAPABILITIES OF THE COMPUTER MODEL

The original purpose of developing the CELCAP code was to evaluate the performance of cogeneration energy systems. There are several key features built into the code that provide it with much broader capabilities. One of these key features is the ability of the CELCAP code to evaluate cogeneration systems with up to five engines of four different types. The second key feature provided in the code is the ability to evaluate the system with the engines operating in three diff t control modes. Apart from these two key features, there are several que features such as the capacity to vary the energy prices, escalat electric utility rates, and engine ratings. This permits evaluat the performance for different price/system scenarios.

The CELCA plications can be broadly classified into the following three groups:

(1) Evaluation of existing power plants with or without cogeneration systems.

(2) Evaluation of modifications and additions to the existing cogeneration systems.

(3) Evaluation of several cogeneration options in the selection of a new system.

In the case of existing cogeneration energy systems, the CELCAP code can be used to study the impact of energy price/escalation rates on the system. Operation of the engines in the existing system in different control modes can be studied to determine the best method of operation in terms of operating costs. This type of study is relevant and meaningful for cogeneration systems with several engines. The impact of changing electrical and steam loads on the system performance can be evaluated.

In the second type of application with the CELCAP code, the modifications and additions to an existing cogeneration system can be evaluated. These include changes in the existing system such as change in the number or type of engines in the system, change in the power plant rating, and change in the electrical and steam loads for the modified system. Such an analysis by CELCAP provides information on the performance of the modified system that could be used in the decisions on modification of the actual system.

In applications involving the selection of new cogeneration energy systems, the CELCAP code can be effectively used to screen several options. Because the cost and time needed to run the code are very little, it is easy to study several arrangements of engine capacities and numbers with the code. With the built-in ability of the code to evaluate the performance for three engine control modes, the selection process for the new system can take into consideration both the system configurations and the manner of operation of the system that meets the desired economic criteria.

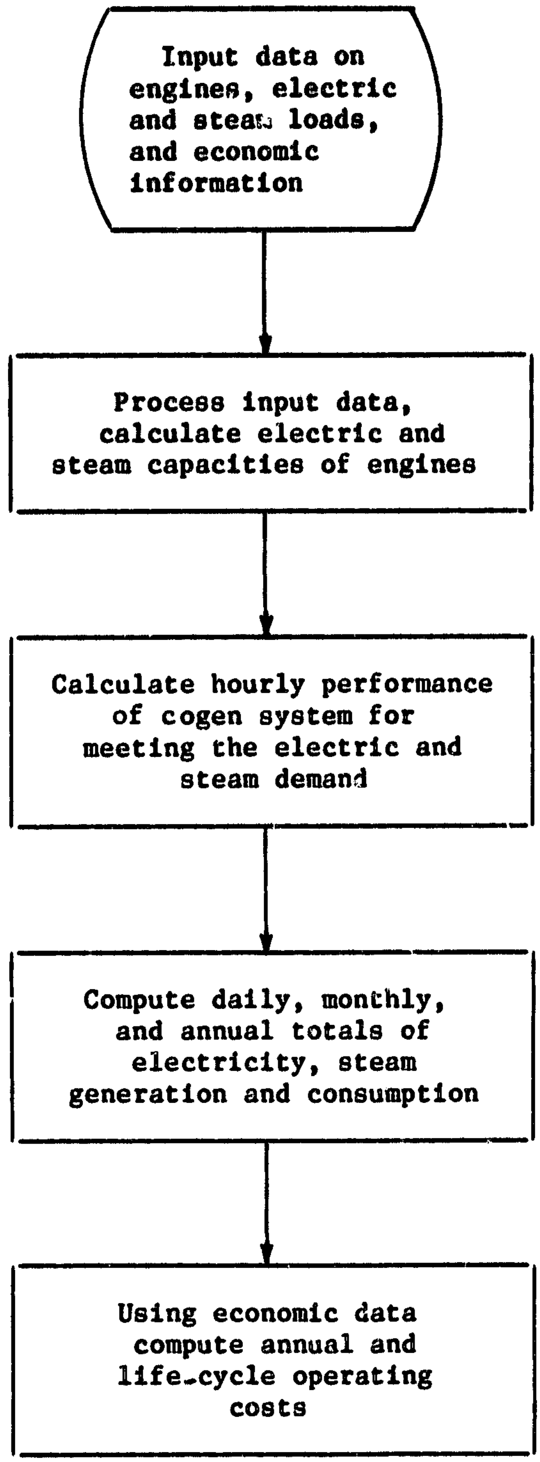

Figure 1-1. Overall Flow Diagram of the CELCAP Code

SECTION II

METHODOLOGY

A. COGENERATION SYSTEM MODEL

A cogeneration energy system is a power plant located on the user's facility that provides electric and thermal energy to the user. To ensure that all the electric and thermal needs of the user are met all the time, it is always connected to an electric or thermal grid and also has auxiliary boilers for producing steam, or auxiliary engine/generators for producing electricity. This is a broad definition of the cogeneration system and covers all types of cogeneration plants that are in existence. However, one can find in practice cogeneration plants that may have just one aspect of this description of the cogeneration system. It is quite common to find cogeneration plants that generate only a part of the electric needs of the user and buy the rest from the utility all the time. The cogeneration model used in the CELCAP code and described in this report consists of a power plant with auxiliary boilers that is located on the user's facility and connected to the electric grid for buying from or selling to the utility grid a part of its electricity.

A block diagram of the cogeneration system model used in CELCAP is shown in Figure 2-1. In this system model, the power conversion system block represents all the engines and boilers in the cogeneration system. Up to five engines of four different types can be included in the CELCAP model. Four types of engines are modeled in CELCAP and also modeled is an auxiliary boiler. The electric utility block in the system block diagram represents the outside utility into which the cogeneration system is tied. In the CELCAP model this block consists of information on electric utility rates for buying and selling electric power between the grid and the cogeneration plant. The electric and steam block in the diagram represents the user's load that has to be met by the cogeneration plant.

The engine models included in CELCAP are the thermal models of the various engines used in the cogeneration system. These models essentially relate the fuel consumption rate to the electrical and/or steam output of the prime mover. They are developed in such a way that by merely using broad specifications of the engine, the model computes the performance of the engine. An example of a model is shown in Figure 2-2. A detailed description of the heat-engine models is given in Section III.

The electrical and steam load model represents the actual demand of the user that has to be met by the cogeneration system. The electric and steam demand of the user is continuously changing because of the working or production procedures of the user. The load model used in the cogeneration system model consists of the hourly data on the electric and steam demand of the user. The load model used in the cogeneration system model uses hourly electric and steam load profiles for a working day and a weekend day for each month of the year. A detailed description of the load model is given in Section IV.

B. COGENERATION SYSTEM SELECTION AND SIZING

Selection of a cogeneration energy system for a specific application is a complex task. First, the availability of many types of prime movers, along with the use of auxiliary boilers, results in many arrangements of the system that could be used for the application. The possibility of buying from or selling to the utility a part of the electrical power of the cogeneration system increases the difficulty in selecting the size of the system. Because of this, it is essential that an analyst have a good knowledge of the equipment that goes into making up the system and the actual application for which the system will be used. The most important criterion in the selection of a cogeneration system for an application is how cheaply and efficiently the energy can be supplied to the user's application. However, there are other key criteria to be taken into account during the system selection process. Some of these are: (1) how easily the system could be modified in the future, (2) how the changes in the electric and steam demand of the user would affect the system operating cost, and (3) how changes in the electric/steam ratio would affect the cost and performance of the system.

In the selection of the system, it is clear from the criteria listed that the analyst should have a good knowledge of the potential cogeneration system and the application itself. Apart from this, he should have a good idea of the future growth of the user and the changes in the future price of purchased fuel and electricity. The CELCAP code can be used to obtain answers to many of these questions. However, it is important that all these questions be answered and data on the potential candidates be collected before the CELCAP code is actually used. This not only lessens the effort involved in the selection process but also ensures that a large number of cogeneration systems are considered for the application in the selection process.

The description of the detailed steps in the selection procedure is outside the scope of this report. In Section VI, a few case studies using CELCAP are described and they provide some of the description of the selection procedure. In a report on DEUS computer evaluation model, Anand, et al. (Reference 10) describe the procedures for system sizing for an industrial process application. It is clear to the author (see Reference 11) that previous experience in cogeneration system selection helps a great deal in the system selection process.

The first step in the selection of a cogeneration system is to make a list of potential candidates for the system. The candidates should consist of single or multiple combinations of the four types of engines described earlier. Because the maximum number of the engines a system can have for evaluation by CELCAP is five, this number should not be exceeded. Some of the candidates can easily be eliminated from evaluation on the basis of limitations on plant size. Diesel systems are the most economical at plant sizes of ~ 200 kW or below, whereas for very large (several megawatts) plant sizes, gas and steam turbines are more attractive. Sometimes it is more economical to have two 500-kW gas turbines than one 1000-kW gas turbine because of the way electric and

thermal loads exist at the plant. Therefore, a large list of potential candidates with wide ranging combinations of the types and number of engines should be developed.

Once a list of potential cogeneration systems is made, the next step in the selection process is to use the CELCAP code. Each of the candidates is examined for several modes of operation and different fuel and purchased electricity prices and escalation rates. The final selection of the cogeneration system is made after a careful examination of the data obtained from the CELCAP code.

C. LOAD ANALYSIS

The function of a cogeneration system is to supply electrical and thermal energy to the facility where it is installed. Therefore, an evaluation of a cogeneration system is always made in consideration of the user being served by the system. The system sees the user as requiring supply to a continuously varying electrical and thermal load. Once the cogeneration system is designed and installed, it meets the demands of the user by operating its engines and boilers at varying capacity levels and buying from or selling to the utility company varying amounts of electricity. The performance of the system is determined by how it operates through the year meeting the energy requirements of the user. Therefore, the electrical and thermal loads of the user to be supplied by the system affect the system performance. It is important, in the cogeneration system evaluation, that the electrical and thermal loads of the user be carefully analyzed so that they accurately represent the user.

The evaluation of the cogeneration system is performed on an annual basis. This requires that data on the electrical and thermal demands of the user be available for at least 1 year. Usually, the electrical and thermal demands of a user vary during the year because of the changing working and weather conditions. However, for the evaluation purposes, assuming the working conditions of the user remain the same, it is safe to assume that the user's energy-demand pattern does not change significantly from year to year. In general, the energy demand of the user is continuously changing. For the type of users considered for the cogeneration application, experience indicates that an hourly representation of the demand is quite accurate for evaluating the system performance.

The information available on the energy demand of a user varies significantly, depending on the type of user. There are users who have all of their energy-using equipment fitted with instruments to record their demand on a continuous basis. Other users have very little information on their energy demand except for boiler capacities and monthly fuel and electric energy consumption figures. While evaluating a cogeneration system for a particular user, it is important to determine how detailed the user's load information should be.

It is generally accepted by the engineers working in the field that hourly profiles of the energy demand of the user provide the required information quite thoroughly. The next question faced by the engineer is whether he should provide the hourly profile of the load for each day of the year. This leads us to examine why the loads of a user change.

The electric and thermal energy loads of an user change primarily because of two factors: (1) change in working conditions, and (2) change in weather conditions. The working conditions determine the use of several types of equipment on the user's facility. Some of the end uses of this equipment are air conditioning, lighting, industrial processing, cooking, and washing. Based on working conditions, the days in a year can be classified into working days and non-working days. Non-working days are the weekend days and holidays when none of the regular work takes place. The loads during the working day itself change because of working and non-working hours. Since an hourly profile for the whole day (24 hours) is provided for the system evaluation, one need not be concerned about the working and non-working hours during the day.

Changing weather conditions also affect the electric and thermal load of the user. Some of the energy end uses that are affected by the weather include air conditioning and water heating. The weather change during the day is automatically taken care of because of the use of the hourly load profile. The weather pattern usually does not change significantly from one day to the next. However, it changes gradually, and over a period of weeks or months it will have changed significantly. In fact, the extreme weather patterns for any location occur usually about 6 months apart, the warmest being during the peak summer day and the coldest during the peak winter day. Because of this, it will not be too inaccurate if the load profile of one chosen working day is used to represent the several working days before the chosen day. It is shown by engineers working on cogeneration system evaluation that one hourly profile of a day can be used to represent all the days in the month. However, separate profiles have to be used for working days and non-working days. The usual practice is to use two hourly load profiles for each month of the year, one for a working day and the other for a non-working day.

To develop 1-year load data for the user, it is necessary to examine the data for several years. This would ensure that the load profiles developed would represent the average weather conditions that can be expected at the user's location. There are several sources of information available about the user's load data that can be used for developing the load profiles for an average year. These include logs maintained at the on-site power plant on hourly electricity generation and purchases, steam generation and export, and daily, monthly, and annual fuel consumption; weather data listing hourly ambient temperatures, daily maximum and minimum temperatures, daily and monthly heating/cooling degree days; data on the capacities and duty cycle of various energy-using equipment; data on occupancy rate and profiles of the buildings on the user's premises.

The first step in developing the average load profiles of the user is to estimate the electric and steam consumption of the user for an average weather condition at the site. Because the load profiles are developed for each month, the average energy consumption and the weather conditions are computed on a monthly basis. To do this, the monthly electric and steam consumption figures for the user are plotted against the corresponding monthly heating/cooling degree days. The data for several of the previous years are used for this purpose. Any change in the total capacity of energy-using equipment should be accounted for while computing the monthly energy consumption figures of the previous years. Separate straight lines are drawn for the heating degree days and cooling degree days that would best fit all the data points. A typical plot of such data is shown in Figure 2-3. Now, using the average degree day numbers for each month of the year, the electric and steam consumption for that month is read from the plot.

The second step is to develop the actual hourly load profiles for working- and non-working days for each month of the year. To do this, the actual available hourly load profiles of the working days for the month are examined and the best representative profile is chosen. A similar procedure is also followed to choose a load profile for the non-working day of the month. Once the load profiles for the working and non-working days of the month are chosen, the next step is to make sure that they agree with the monthly energy figures. The monthly energy figure represented by the load profiles is given by the sum of the two products: number of working days in the month times area under the working day profile, and number of non-working days in the month times the area under the non-working day profile. This sum may not be equal to the monthly energy consumption picked from the monthly energy consumption versus degree-day plot. In such a case, the multiplying factor should be calculated as follows:

$$K = E_m/(AE_{wj} + AE_{nwj})$$

where

- K = multiplying factor
- E_{mj} = average monthly electrical energy consumption of the user for the month j selected from the monthly electricity consumption versus degree day plot
- AE_{wj} = area under the typical working day hourly electric demand profile of the user for the month j
- AE_{nwj} = area under the typical non-working day hourly electric demands profile of the user for the month j

Each hourly load in the load profiles is multiplied by this factor to obtain the correct working- and non-working-day load profiles for the month. This procedure is repeated for all the months of the year for both electric and steam demand profiles. The resulting load profiles represent the load demand of the user for average weather conditions at his location.

D. MODES OF OPERATION

Several of the user's features are taken into consideration while designing a cogeneration system. These features include the electrical and steam demand of the user, fuel and purchased electricity prices, and the weather at the user's location. One of the key characteristics of a cogeneration system is that, apart from the prime movers, the system has auxiliary boilers and can sell or buy electricity from the electric utility company. The prime movers in the system can also be modulated so that their electric-to-steam generation ratio can be altered. These features provide the system with a flexibility to operate in several different modes. Once the system is installed, it operates in any one of these modes and supplies electricity and steam to meet the demand of the user.

There are three modes in which the cogeneration system can be operated. In the first mode of operation the prime movers of the cogeneration system are operated at their full-rated capacity. All the electricity and steam from the prime movers are used to supply the user's load. Any excess electrical output is sold to the utility company and any excess steam output is thrown away. If there is a shortfall in electrical output, it is made up by that purchased from the utility. Similarly, any shortfall in meeting the steam demand by the steam from the prime movers is made up by an auxiliary boiler. In the second mode of operation, the prime movers in the system are operated to meet all the electrical load demand of the user. The steam output of the prime mover is used for meeting the steam demand of the user. If there is any shortfall between the demand and the output, it is made up by an auxiliary boiler. In case the steam output is in excess of the demand, the excess amount is thrown away. In the third mode of operation the prime movers in the cogeneration system are operated so that they put out enough steam to meet all the demand of the user. In this mode, any shortfall in meeting the user's electric demand is made up by the purchased electricity. Similarly, if there is any excess electric supply after meeting the demand, it is sold to the utility company.

The prime movers used in a cogeneration system considered in this report are the steam turbine, gas turbine, and diesel engine. For the gas turbine and diesel engine, if the engine electric output is given, the amount of steam that can be generated by the engine is fixed. Whereas in the case of the auto-extraction turbine and back-pressure steam turbine, for a given electric output of the turbine, there is a wide range in the amount of steam that is available from the turbine. In Figure 2-4 a performance map of the automatic-extraction turbine is shown. It can be seen that for a given engine electric output there is a wide range of steam extraction rates possible. What this means is that these two types of steam turbines can meet both the electric and steam demand of the user without using an auxiliary boiler or purchasing electricity from the utility company. This is one of the reasons that makes an auto-extraction steam turbine and back-pressure steam turbine attractive candidates as cogeneration system prime movers.

E. ECONOMIC EVALUATION

The economic evaluation is an important part in the overall evaluation of the cogeneration system. The economic evaluation of the system not only provides the cost of installing and operating the system, but also helps in comparing competing cogeneration systems. Because one of the criteria in the selection of a cogeneration system is how cheaply the system can supply the energy to the user, an economic evaluation of all the cogeneration systems that are suitable for the specific user is essential in the system selection process.

Basically, the economic evaluation of a cogeneration system consists of calculating the total annual cost of supplying the electrical and thermal demands of the user. The annual cost is made up of several basic contributing factors. These factors include capital cost expenditures, fuel costs, operation and maintenance costs, and purchased energy costs. The costs for future years have to be estimated because the energy prices and operation and maintenance costs are unknown. For this purpose, escalation rates for fuel and O&M costs are estimated. The results of the economic evaluation are the annual and life-cycle operating costs.

The economic evaluation methodology used in the CELCAP code is described by Cooper (see Reference 1) in his report on cogeneration systems. In this evaluation, cost comparisons are made of the total cost of providing both thermal and electrical energy. This total annual cost is expressed as follows:

$$TC_j\ (Y) = CC_j\ (Y) + F_j\ (y) + OM_j\ (y) + P_j\ (y) - R_j\ (y)$$

where

y = the year for which the annual costs are computed

j = costs if alternative "j" is chosen to supply the energy to the user

TC = total cost for thermal and electrical services

CC = capital cost expenditure including interest on funds during construction

F = fuel costs

OM = operation and maintenance costs

P = cost of payment for energy purchased from outside (electrical or thermal) or for services purchased from outside

R = any revenues resulting from operation and/or ownership of equipment at the user's facility.

All the costs are in actual dollars. Cost estimates for future years, y, are calculated by escalating the current costs at the rates assumed appropriate for them.

The total annual costs are calculated for various alternatives. These may be variations in design options, ownership/operation arrangements, or financial arrangements. The different design options may include different types of engines or different engine capacities. One alternative may reduce fuel costs and increase capital costs. Another may involve a larger plant that increases both fuel and capital costs, but results in revenues through sale of excess power to offset the increases. The ownership/operation of the plant may involve several alternatives. If coordinated with the local utility company, the following three alternatives may be possible: (1) the local utility will own and operate the entire cogeneration plant and sell both steam and electrical power to meet the user needs; (2) the local utility company may own and operate only the electrical generation portion of the cogeneration plant, with provisions for extracting steam needed for the user, and provide electrical service to the user; or (3) the utility company will not own or operate any part of the plant, but will sell power to the user as needed or buy excess power from the cogeneration plant. Based on the financial arrangements, there will be several alternatives, for example, various means of financing the construction costs.

In the economic evaluation, the cogeneration alternatives are compared to a baseline system. The baseline system is usually the system that is already in existence or a system where on-site boilers supply the steam to the user and all the electricity is purchased from the utility. If the cogeneration options are being compared to the existing system, the capital cost of the existing system will be zero, except for the year when a new piece of equipment is purchased. The total annual cost for the existing system and the cogeneration alternative is given by

$$TC_{bl}(y) = CC_{bl}(y) + F_{bl}(y) + OM_{bl} + P_{bl}(y) - R_{bl}(y)$$

$$TC_j(y) = CC_j(y) + F_j(y) + OM_j(y) + P_j(y) - R_j(y)$$

The subscript, bl, refers to the baseline or the existing system and all the other terms remain the same as described earlier. The annual savings to the owner/operator from this is given by

$$S_j(y) = [TC_{bl}(y) - TC_j(y)] - [CC_{bl}(y) - CC_j(y)]$$

The measure used in judging the economic viability of the investment in a system is Return on Investment (ROI). A minimum ROI must be exceeded to approve the investment. This miniumum ROI is set by the investor himself. The acceptable ROI is a function of the economic life for the investment, which is also established by the investor. For an investment in a candidate to be

profitable, the total savings should exceed the net capital expenditure. Substituting the minimum ROI for the savings,

$$\sum_{y = N+1}^{y = EL + N} S_n/(1 + ROI_j)^y > \sum_{y=0}^{y = EL + N} (CC_j(y) - CC_{bl}(y))/(1 + ROI)^y$$

where

$CC_j(y)$ = annual construction payments made for designated project

N = number of years from beginning of construction financing to start-up of alternative system

EL = economic life of alternative system

S_j = annual savings from operating the system

If the baseline system is retained, there will not be any capital expenditure. This means that $CC_{bl}(y)$, the capital cost for the baseline system, is zero for all years through the economic life of the alternative system. Then the expression for the investment in a system to be viable is

$$\sum_{y = N+1}^{y = EL + N} S_j\ (y)/(1 + ROI_j)^y = \sum_{y=0}^{y = N} CC_j(y)/(1 + ROI_j)^y$$

The measure ROI is similar to the discount rate that the Navy uses to convert future savings or expenditures into present values. As used by the Navy, the discount rate is considered to be the rate of return over and above the inflation rate. Therefore, in the escalation rates of fuel costs, O&M costs are considered as price increases over and above those required to keep up with inflation. Utility companies or other firms often choose rather to include inflation in their ROI, and appropriately account for inflation in other terms also.

The Navy uses the measure discounted Savings to Investment Ratio (SIR) to assess the viability of a project. This SIR is obtained by setting a minimum ROI and calculating the ratio of the left-hand side (savings) of the above equation to the right-hand side (investment). The Navy uses the figure of 0.10 for ROI. In general, for a viable investment this ratio should be greater than 1.

$$SIR = [\sum_{y = N+1}^{y = EL + N} S_j(y)/(1 + ROI_j)^y]/[\sum_{y=0}^{y = EL + N} (CC_j(y) - CC_{bl}(y))/(1 + ROI)^y] > 1$$

Not all alternatives involve capital investments. In such cases, ROI cannot be used. However, in these cases, the declining value of money is accounted for by discounting the future expenditures. The Navy chooses to discount 10% over inflation in all cases, but other businesses might elect to use a different discount rate where capital expenditure is not involved. Using the relation for ROI and substituting for $S_j(y)$, it can be shown that an alternative system is viable when its discounted total life-cycle cost is less than that of the baseline approach for operating the system

$$\sum_{y=0}^{y=EL+N} TC_{b1}(y)/(1+d_j)^y > \sum_{y=0}^{y=EL+N} TC_j(y)/(1+d_j)^y$$

where d_j = discount factor used for the specific type of expenditure involved.

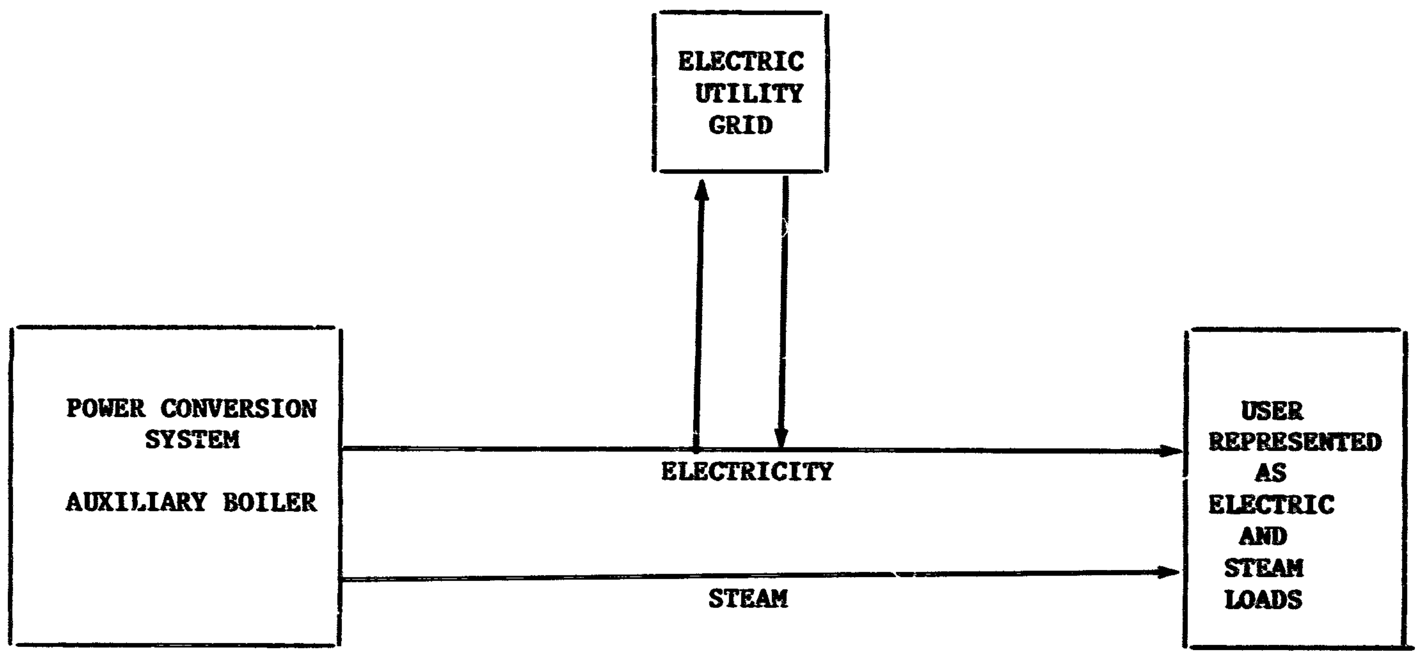

Figure 2-1. Cogeneration System Model Used in CELCAP

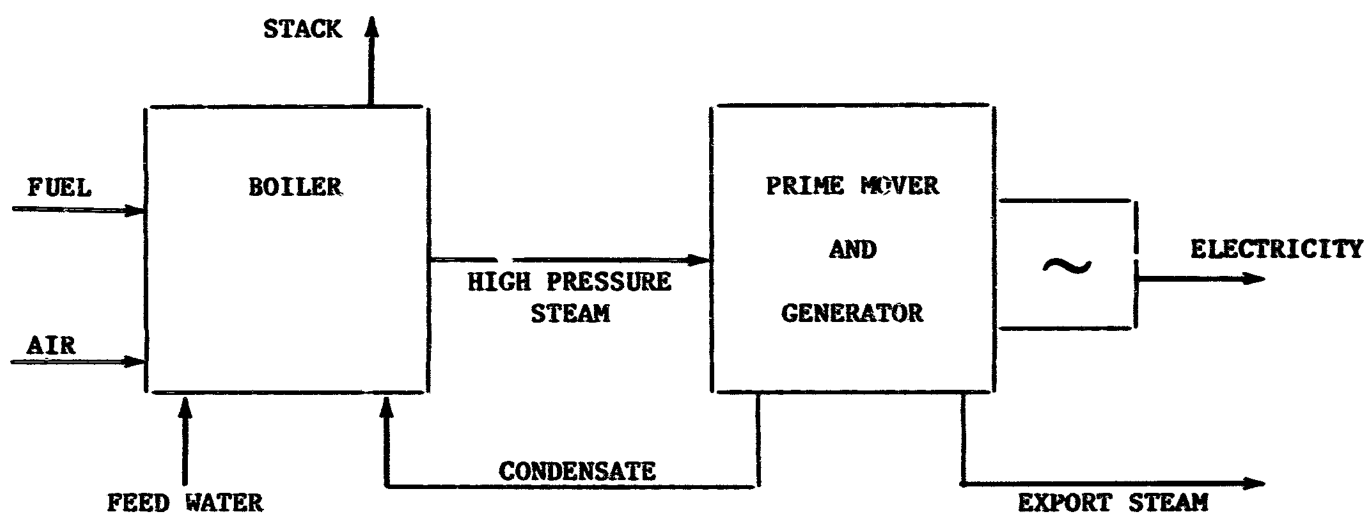

Figure 2-2. Typical Heat Engine Model Used in CELCAP

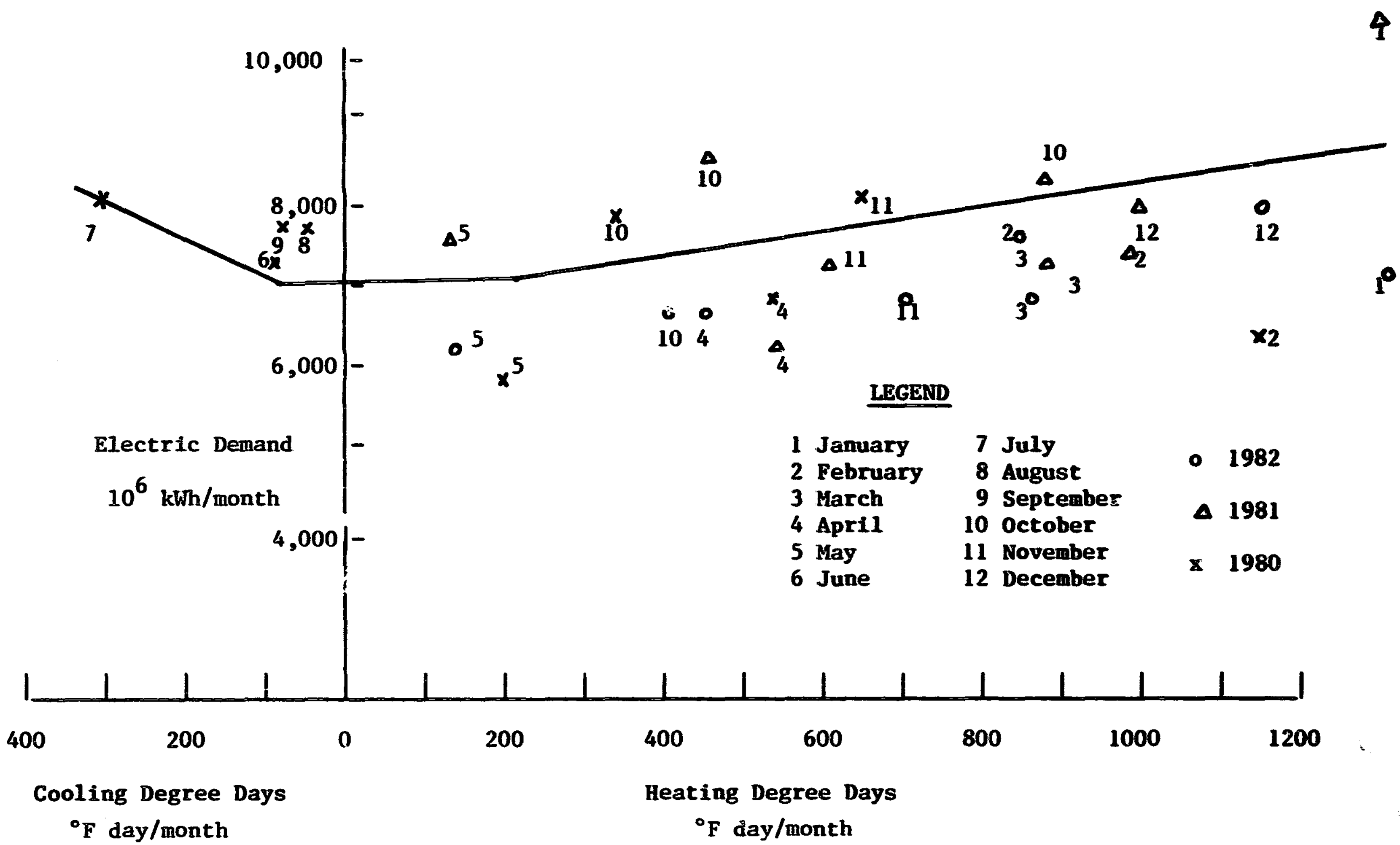

Figure 2-3. Monthly Electrical Energy Consumption as a Function of Heating and Cooling Degree Days

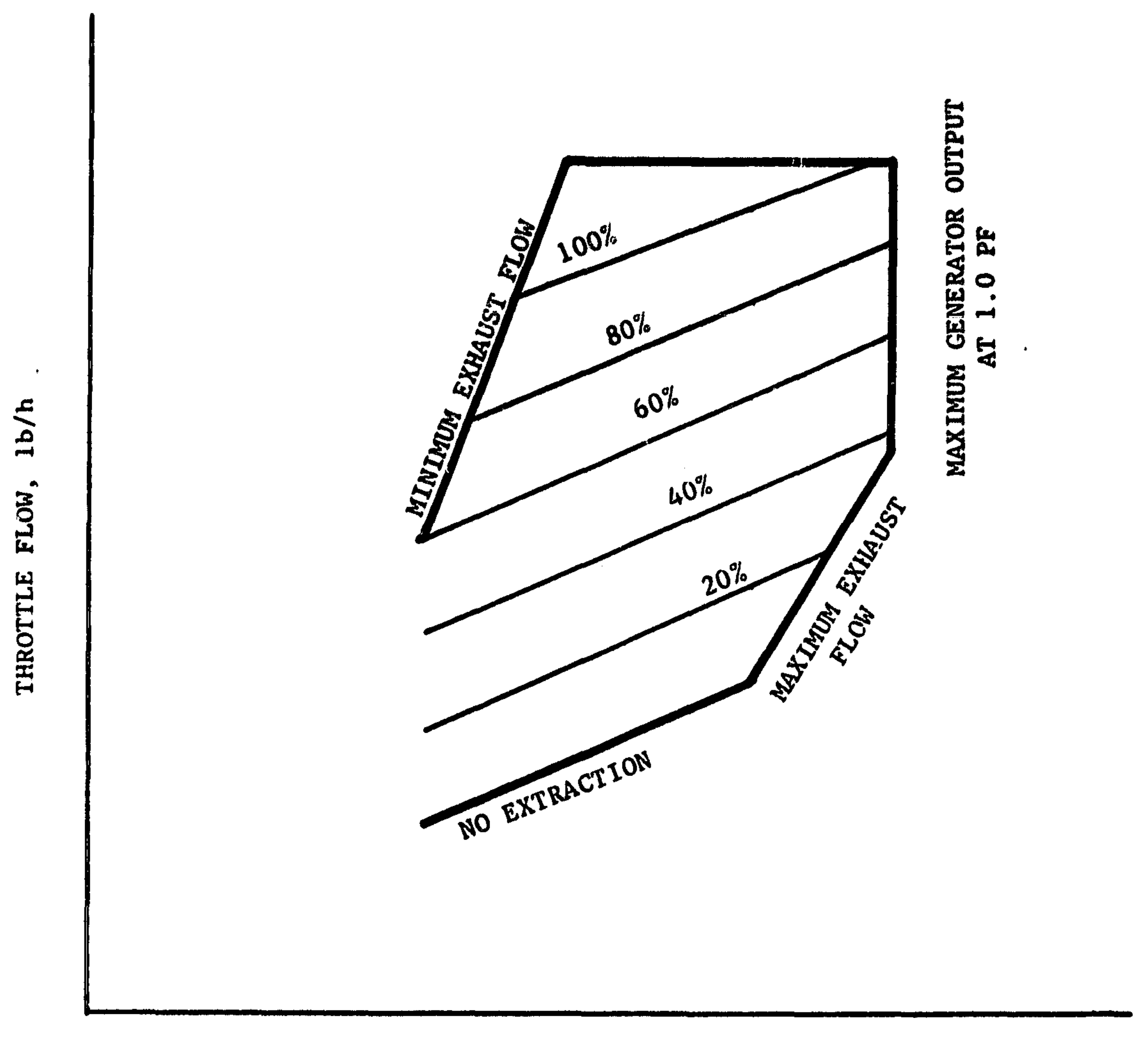

Figure 2-4. Typical Performance Map of a Single-Automatic Extraction Turbine

SECTION III

HEAT ENGINE MODELS USED IN CELCAP

In the CELCAP code four different types of engines are modeled. These four engines are the gas turbine with exhaust heat boiler, the diesel engine with exhaust heat boiler, the automatic-extraction steam turbine, and the back-pressure steam turbine. A detailed description of the thermodynamic relations used in developing these models is presented in this section. These models, through their relationship, basically compute the fuel consumption rate for generating the desired electrical power.

A. GAS TURBINE WITH EXHAUST HEAT BOILER

The gas turbine with an exhaust heat boiler is one of the four cogeneration engine systems modeled in the CELCAP code. In this system, an open-cycle gas turbine drives an electrical generator and the exhaust gases from the turbine pass through a boiler where some of the heat in the exhaust gases is recovered for generating steam. A schematic of this arrangement is shown in Figure 3-1. The models of the gas turbine and the exhaust boiler are integrated to obtain the system model.

The thermodynamic cycle on which the gas turbine operates is a Brayton cycle. The type of gas turbine modeled in the CELCAP code is the open-cycle turbine. This cycle consists of a compression process; a constant-pressure, heat-addition process; an expansion process; and a constant-pressure, heat-rejection process. The working fluid, which is air in this case, first passes through the compressor where its pressure is increased. The high-pressure air then passes through the combustion chamber where the fuel is injected and the heat of this combustion increases the gas temperature. The hot gases, which are at high pressure and temperature, are expanded in the turbine. The expanded gases, which are at ambient pressure, are generally exhausted to the atmosphere. However, for the gas turbine system with an exhaust heat boiler, the exhaust gases pass through a boiler before they leave the system.

The basic relationships used in the gas turbine model are

$$\dot{W}_C = \dot{M}_{AIR} * C_{PCMPR} \; (T_2 - T_{AMB}) \qquad (1)$$

$$\dot{Q}_F = (\dot{M}_{AIR} + \dot{M}_F) \; C_{PCOMB} \; (T_3 - T_2) \qquad (2)$$

$$\dot{Q}_F = \dot{M}_F (HV) \qquad (3)$$

$$\dot{W}_T = (\dot{M}_{AIR} + \dot{M}_F) \; C_{PTRBN} \; (T_3 - T_{EXH}) \qquad (4)$$

$$\dot{E} = (\eta_G/3414) \; (\dot{W}_T - \dot{W}_C) \qquad (5)$$

where

$\dot{W}_C$ = compressor work, Btu/h

$\dot{M}_{AIR}$ = mass flow rate of air, lbm/h

C_{PCMPR} = specific heat of air through the compressor, Btu/lbm-°R

T_2 = air temperature at outlet of the compressor, °R

T_{AMB} = ambient air temperature, °R

$\dot{Q}_F$ = fuel flow rate, Btu/h

$\dot{M}_F$ = fuel flow rate, lbm/h

C_{PCOMB} = specific heat of combustion gases, °R

T_3 = turbine inlet temperature, °R

H_V = lower heating value of fuel, Btu/lbm

$\dot{W}_T$ = shaft power, Btu/h

C_{PTRBN} = specific heat of the gases passing through the turbine, Btu/lbm-°R

T_{EXM} = exhaust gas temperature, °R

$\dot{E}$ = electrical power generated, kW

η_G = efficiency of the generator (decimal)

The performance data for industrial turbines is usually given at their design conditions. These data consist of the ambient temperature, T_{AMBD}; mass flow rate of air, M_{AIRD}; generator output, $\dot{E}_D$; and the fuel flow rate, $\dot{Q}_{FD}$. Also provided to the model is the part-load performance of the turbine in the form

$$\dot{Q}_F/\dot{Q}_{FD} = f(\dot{E}/\dot{E}_D)$$

However, to calculate the part-load performance of the gas turbine-exhaust heat boiler system, more information is needed on the internal conditions in the turbine. For this reason, several relationships were developed (see Reference 1) so that an analysis procedure can be applied with the performance data.

The first of these is for relating the off-design compressor work to compressor work at the design conditions. The turbine-generator set rotates at a constant rpm to generate power at constant frequency. Because of this, the volume flow rate of air that is compressed remains constant, regardless of ambient conditions or electrical power output. The compression ratio in the compressor is also constant. Reference 1 shows that the compressor work, $\dot{W}_C$,

at off-design condition is related to the work at design conditions as follows

$$\dot{W}_C = \dot{W}_{CD} * P_{AMB}/P_{AMBD} \tag{6}$$

$T3_{LIM}$, the limiting turbine inlet temperature, usually set to $T3_D$, the design turbine inlet temperature. This limit is used to establish the maximum engine/generator output power at off-design conditions.

The second of these relationships is developed for computing compressor work at design conditions in terms of the input heat rate to the turbine. Reference 1 shows that the compressor work can be computed as follows:

$$\dot{W}_{CD} = (C_{PTRBN}/C_{PCOMB}) * \dot{Q}_{FO} \tag{7}$$

where $\dot{Q}_{FO}$ = the fuel flow rate at the turbine idle condition when the output of the turbine is zero. The specific heat term is included because even small changes in the specific heat of the working fluid can have significant effect on the predicted performance of the turbine.

The third relationship used in the analysis procedure concerns the turbine inlet temperature for continuous operation of the turbine. At off-design conditions, the turbine inlet temperature, T_3, is not allowed to exceed some maximum value, $T3_{LIM}$, usually set to $T3_D$, the design turbine inlet temperature. This limit is used in establishing the maximum engine/generator output power at off-design conditions.

A schematic of the exhaust boiler is shown in Figure 3-1 along with that of the gas turbine. The boiler usually consists of three sections: a superheater, an evaporator, and an economizer. Depending on whether the boiler produces superheated steam, saturated steam, or hot water, one or more of the sections may not be present.

The exhaust gases from the turbine enter the boiler at the superheater section and pass through the evaporator and the economizer sections. The temperature profile of the exhaust gases and water/steam as they pass through the boiler is shown in Figure 3-2. The hot exhaust gas is cooled from T_{EXH} as it passes through the superheater and evaporator sections and gives up heat to the evaporating steam. The temperature of the gas leaving the evaporator section is designated as the "pinch point" temperature, T_{PINCH}. The gas drops its temperature further in the economizer by giving up heat to the boiler feed water. In the economizer, the boiler feed water temperature increases from the de-aerator exit temperature, T_{BLFRD}, up to evaporator temperature corresponding to the steam pressure, T_{EVP}. Preheating and de-aerating are accomplished outside the boiler. Plant steam may be used for this where the water temperature

is raised from T_{FW} to T_{BLRFD}.

The mass balance for steam across the boundaries of the boiler is given by

$$\dot{M}_{EXP} = \dot{M}_{GEN} - \dot{M}_{PLNT} = \dot{M}_{FW} - \dot{M}_{PLNT} \quad (8)$$

where

$\dot{M}_{EXP}$ = steam exported out of the power plant, lbm/h

$\dot{M}_{GEN}$ = steam generated at the power plant, lbm/h

$\dot{M}_{PLNT}$ = steam used for in-plant use, lbm/h

As for the mass balance for the exhaust gases, all the exhaust gas from the turbine passes through the exhaust heat boiler. The only exception is when there is small leakage of flow from the ducts between the turbine and the boiler. A factor, K, is introduced to account for flow loss between turbine and the boiler or the intentional diversion of a portion of the turbine exhaust flow. The factor is defined as

$$K = \dot{M}_{BLR}/(\dot{M}_{AIR} + \dot{M}_F)$$

where

$\dot{M}_{BLR}$ = flow rate gas entering the boiler, lbm/h

$\dot{M}_{AIR}$ = flow rate of air entering the compressor, lbm/h

$\dot{M}_F$ = flow rate of the fuel injected to the combustor, lbm/h

The value of K, at best, is about 0.98. There may be cases where the flow of gas through the boiler is modulated by a valve in the duct between the turbine and the boiler as means of controlling the steam production. In such cases, K indicates the fraction of available turbine exhaust flow that is directed through the exhaust heat boiler.

For the superheater and evaporator sections, four expressions are written relating heat transfer between the exhaust gas and the steam. The first of these relationships is the heat loss from the exhaust gas given by

$$\dot{Q}_{STM} = K(\dot{M}_{AIR} + \dot{M}_F)\ C_{PBLR}\ (T_{EXH} - T_{PINCH}) \quad (9)$$

The second relationship is for the heat gained by the water given by

$$\dot{Q}_{STM} = \dot{M}_{GEN} (h_{STM} - h_{SL}) \tag{10}$$

where

h_{STM} = enthalpy of steam, Btu/lb

h_{SL} = enthalpy of saturated liquid, Btu/lb

The third relationship is for the heat transferred between the gas and water and is given by

$$\begin{aligned} \dot{Q}_{STM} &= UA\ DT_{LM} \\ &= UA * (T_{EXH} - T_{STM}) - (T_{PINCH} - T_{EVP})/ \\ &\quad \ln [(T_{EXH} - T_{STM})/(T_{PINCH} - T_{EVP})] \end{aligned} \tag{11}$$

UA is the product of the overall coefficient of heat transfer and the heat transfer area for the superheater and evaporator sections. For a given boiler design, the area, A, does not change. Even though the factor, U, changes with the flow conditions, specifically with conditions that change the Reynolds number of flow around the tubes, the rate of change of U with flow conditions is not rapid. Therefore, it is a reasonable assumption that the product UA is approximately constant, and UA = $(UA)_d$.

The fourth relationship expresses the heat recovery boiler effectiveness in terms of evaporation temperature and the design exhaust and pinch temperatures. The amount of steam that can be generated is limited by the fact that the gas temperature cannot drop below the evaporation temperature, T_{EVP}. The boiler heat recovery effectiveness is given by

$$e = (T_{EXHD} - T_{PINCH})/(T_{EXHD} - T_{EVP}) \tag{12}$$

The effectiveness, e, is increased by increasing the heat transfer surface area. The heat transfer area required to reach an effectiveness value of unity increases logarithmically. Usually there is a point where the improved effectiveness does not justify the additional material costs required for the heat transfer area. A value of e = 0.92 is considered as the maximum economically achievable effectiveness for most applications.

For the economizer, there are two governing equations involving mass and energy balance. The energy balance states that heat given up by the exhaust gas equals the heat gain by the water in raising its enthalpy from h_{BLRFD} to h_{EVP}.

$$K(M_{AIR} + \dot{M}_F)\, C_{PBLR}\, (T_{PINCH} - T_{STK}) = (\dot{M}_{FW} + \dot{M}_{PLNT})(h_{EVP} - h_{BLFRD}) \qquad (13)$$

The mass balance for the economizer is given by

$$\dot{M}_{BLDN} = L * \dot{M}_{GEN} \qquad (14)$$

where L is the fraction of generated steam that accounts for blowdown.

In the present analysis, it is assumed that the feedwater is mixed with and heated by condensing steam from the evaporator/superheater section of the boiler. The heated de-aerated water emerges at temperature T_{BLRFD}. Neglecting any mass loss due to outgassing and loss of flashing steam, the energy balance is given by

$$\dot{M}_{FW} * I * h_{FW} + \dot{M}_{PLNT} * h_{STM} = (\dot{M}_{FW} + \dot{M}_{PLNT})\, h_{BLFRD} \qquad (15)$$

All the equations presented for the boiler so far apply only to unfired exhaust boilers. When a flame is present in a boiler, a portion of the transfer of heat from the hot gases to the tube walls is radiative. The equations presented do not account for this. In the CELCAP code, when the export steam from the boiler is insufficient to meet the load, it is assumed that the additional steam requirement is met by a completely separate boiler.

The data that is provided to the gas turbine model include the turbine and boiler design values and the atmospheric conditions. The turbine design values are:

T_{AMBD} = ambient temperature, R

P_{AMBD} = ambient pressure, lb/sq in.

$\dot{M}_{AIRD}$ = mass flow rate of air into the gas turbine, lb/h

$\dot{E}_D$ = generator power output at full load, kW

$\dot{Q}_{FD}$ = fuel consumption at full load, Btu/h

Along with these, a part-load performance curve in the form of $\dot{Q}_F/\dot{Q}_{FD} = f(\dot{E}/\dot{E}_D)$ is also given.

The boiler design values are

h_{STM} = enthalpy of steam, Btu/lb

h_{SL} = enthalpy of saturated liquid, Btu/lb

h_{BLRFD} = enthalpy of water at boiler inlet, Btu/lb

h_{FW} = enthalpy of feedwater, Btu/lb

e = heat exchange effectiveness of waste heat recovery boiler (decimal)

The data on atmospheric conditions include the monthly average values of maximum and minimum temperatures, and the atmospheric pressures. The value of the electrical power to be generated is also provided to the model.

The first step in the computation of the gas turbine performance is to calculate the compressor work, $\dot{W}_C$. Using the engine performance curve

$$\dot{Q}_F/\dot{Q}_{FD} = f(\dot{E}/\dot{E}_D)$$

the fuel rate for no-power condition, $\dot{Q}_{FO}$ is calculated. From the relation shown in Equation (7) the compressor work is then given by

$$\dot{W}_{CD} = C_{PTRBN} / C_{PCOMB} * \dot{Q}_{FO}$$

The off-design compressor work is determined using Equation (8)

$$\dot{W}_C = \dot{W}_{CD} (P_{AMB}/P_{AMBD})$$

At off-design conditions, the mass flow rate of working fluid through the turbine will be different than the design mass flow rate, $\dot{M}_{AIRD}$. Because the turbine rpm remains constant irrespective of the atmospheric conditions, the volume flow rate through turbine remains constant. The mass flow rate at off-design condition is calculated from

$$\dot{M}_{AIR} = P_{AMB}/P_{AMBD} * T_{AMBD}/T_{AMB} * \dot{M}_{AIRD}$$

$$T_2 = T_{AMBD} + \dot{W}_C/\dot{M}_{AIR} * C_{PCMPR}$$

$$T_{2D} = T_{AMBD} + (\dot{W}_{CD}/\dot{M}_{AIRD} * C_{PCMPR})$$

where T_2 and T_{2D} are the air temperature at the compressor outlet for off-design and design conditions, respectively.

Once the compressor outlet temperature is calculated, the next step is to find what the turbine inlet temperature would be to generate power $\dot{E}$. This is calculated as follows:

$$\dot{Q}_F = \dot{Q}_{FD} * f(\dot{E}/\dot{E}_D)$$

$$\dot{M}_F = \dot{Q}_F/HV$$

$$T_3 = T_2 + \dot{Q}_F/[(\dot{M}_{AIR} + \dot{M}_F) * C_{PCOMB}]$$

To make sure that the turbine inlet temperature does not exceed the limiting temperature, T_{3LIM}, the design turbine inlet temperature is first calculated,

$$\dot{M}_{FD} = \dot{Q}_{FD}/HV$$

$$T_{3D} = T_{2D} + \dot{Q}_{FD}/[(\dot{M}_{AIRD} + \dot{M}_{FD}) * C_{PCOMB}]$$

$$T_{3LIM} = T_{3D}$$

If the turbine inlet temperature, T_3, is greater than the limiting temperature, T_{3LIM}, then T_3 is set to T_{3LIM} and the generator power, $\dot{E}$, is calculated using Equations (2) and (3).

$$\dot{Q}_{FLIM} = (\dot{M}_{AIR} + \dot{Q}_{FLIM}/HV) * C_{PCOMB} * (T_{3LIM} - T_2)$$

Solving explicitly for $\dot{Q}_{FLIM}$,

$$\dot{Q}_{FLIM} = [\dot{M}_{AIR} * C_{PCOMB} * (T_{3LIM} - T_2)]/[1 - (C_{PCOMB}/HV) * (T_{3LIM} - T_2)]$$

$$\dot{E}_{LIM} = \dot{E}_D * f^{-1} (\dot{Q}_{FLIM}/\dot{Q}_{FD})$$

$$\dot{E} = \dot{E}_{LIM}$$

Once the generator output power, $\dot{E}$, and the compressor work, $\dot{W}_C$, are calculated, the next step is to calculate the turbine work at off-design and design conditions, respectively.

$$\dot{W}_F = (3413/\ _G) * \dot{E} + \dot{W}_C$$

$$\dot{W}_{TD} = (3413/\ _G) * \dot{E}_D + \dot{W}_{CD}$$

The engine exhaust temperature, which is same as the turbine exit temperature, is computed using the turbine work for the off-design conditions as follows

$$T_{EXH} = T_3 - \dot{W}/(\dot{M}_{AIR} + \dot{M}_F) * C_{PTRBN}$$

and for the design conditions

$$T_{EXHD} = T_{3D} - \dot{W}_{TD}/(\dot{M}_{AIR} + \dot{M}_{FD}) * C_{PTRBN}$$

With the calculation of the exhaust temperature, all the performance parameters of the engine are calculated. The next step in the analysis procedure is to calculate the heat recovery boiler performance. The design point data are to estimate the heat transfer potential, UA, of the boiler. From Equation (12),

$$T_{PINCHD} = T_{EXHD} - e * (T_{EXHD} - T_{EVP})$$

The value of the heat effectiveness in the above equation is set at 0.92.

From Equations (9) and (11) solving for UA, we get

$$UA = K\ (\dot{M}_{AIRD} + \dot{M}_{FD})\ C_{PBLR}\ (T_{EXHD} - T_{PINCHD})/$$
$$[\{(T_{EXHD} - T_{STM}) - (T_{PINCHD} - T_{EVP})\}/\ln XTD]$$

where $XTD = (T_{EXHD} - T_{STM})/(T_{PINCHD} - T_{EVP})$.

For off-design conditions, the temperature profile of the gas flowing through the boiler will be different from that occurring at design conditions.

Specifically, the pinch-point temperature will be different. From Equations (9) and (11), T_{PINCH} is given by

$$T_{PINCH} = T_{EXH} - UA/[K * C_{PBLR} * (\dot{M}_{AIR} + \dot{M}_F)] * [(T_{EXH} - T_{STM}) - (T_{PINCH} - T_{EVP})/\ln XT]$$

where $XT = (T_{EXH} - T_{STM})/(T_{PINCH} - T_{EVP})$.

Because T_{PINCH} appears on both sides of the equation, the expression is solved for T_{PINCH} iteratively.

The next step in the analysis procedure is to calculate the steam generated by the boiler, M_{GEN}

$$\dot{M}_{GEN} = UA/(h_{STM} - h_{SL}) * [(T_{EXH} - T_{STM}) - (T_{PINCH} - T_{EVP})]/\ln XT$$

The steam exported out of the power plant to the load is less than the generated steam by the amount of steam used for in-plant use. The steam used for in-plant purposes is given by

$$\dot{M}_{PLNT} = (1 + L) * \dot{M}_{GEN} * (h_{BLRFD} - h_{FW})/(h_{STM} - h_{FW})$$

The exported steam is given by

$$\dot{M}_{EXP} = \dot{M}_{GEN} - \dot{M}_{PLNT}$$

B. DIESEL ENGINE WITH WASTE HEAT BOILER

Diesel engines are internal combustion piston engines. They convert the combustion energy of the fuel into mechanical energy. In a conventional diesel power generation system, the output of the engine is the mechanical shaft power that drives an electric generator or a mechanical drive; whereas, in a diesel cogeneration system, along with the shaft output of the engine, a portion of the heat in the exhaust is recovered and used for generating steam or hot water. The model of diesel engine included in CELCAP is that of a diesel cogeneration system.

The fuel energy burnt in a diesel engine is converted into the following energy streams: shaft work, heat in jacket cooling water, exhaust gas energy, and lube oil heat. The relative percentages of these energy streams vary, depending on the percent of the rated full load at which the engine is operating. In Figure 3-13, a plot of these energy streams as a percent of input fuel energy is shown as a function of the percent of rated engine load for a typical diesel engine. Also shown in Figure 3-13 is the exhaust gas temperature as a function of percent of the rated load. When the diesel engine is operating as a cogeneration system, a part of the heat in the exhaust gas is recovered and used for producing steam. If the cogeneration system is producing hot water instead of or along with steam, then the engine jacket cooling water is also used for producing the hot water.

The amount of recoverable heat from the diesel exhaust is calculated from the equation

$$\dot{Q}_{REC} = \dot{M}_A * C_{PA} * (T_{EXH} - T_{STACK})$$

where

$\dot{Q}_{REC}$ = heat recoverable from the exhaust in Btu/h

$\dot{M}_A$ = airflow rate in lb/h

C_{PA} = average specific heat in Btu/lb°F

T_{EXH} = exhaust gas temperatures in °F

T_{STACK} = stack gas temperature in °F

The exhaust gases are not cooled below a certain temperature limit, T_{STACK}, to prevent any condensation in the heat exchanger surface of the exhaust recovery boiler. The lower temperature limit, T_{STACK}, to which the exhaust gases can be cooled is usually set at a value in the 300-350°F range.

The first step in the calculation of $\dot{Q}_{REC}$, the amount of recoverable heat from the exhaust, is to calculate the exhaust gas temperature. The exhaust gas temperature depends on the load level at which the engine is operating. If the engine is not operating at its rated capacity, the exhaust temperature is estimated by using the part-load data. The input part-load data consist of E_{DP}, the part-load level, and T_{EXHP}, the exhaust gas temperature at this part-load level. Using the rated capacity, E_D, and T_{EXHD}, the exhaust temperature at the full-rated capacity, the exhaust gas temperature at any fractional capacity, E_{DFR}, can be estimated by

$$T_{EXHFR} = T_{EXHFD} - [1 - (\dot{E}_{DFR}/\dot{E}_D)]/EXHRT$$

where

$$EXHRT = (1 - \dot{E}_{DP}/\dot{E}_D)/(T_{EXHD} - T_{EXHP})$$

$$\dot{Q}_{REC} = \dot{M}_A C_{PA} (T_{EXHFR} - T_{STACK})$$

If the exhaust gas temperature, T_{EXHFR}, at the fractional load, $\dot{E}_{DFR}$, is below the lower exhaust gas temperature, T_{STACK}, then no heat will be available for producing steam.

The amount of steam that can be generated from the recoverable heat is calculated by

$$\dot{Q}_{STMD} = \dot{Q}_{REC}\ \eta_{EFF}/(h_{STM} - h_{FWTR})$$

where

$\dot{Q}_{STMD}$ = amount of steam produced by the heat recovery boilers, lb/h

η_{EFF} = efficiency of the waste heat recovery boiler

h_{STM} = enthalpy of steam, Btu/lb

h_{FWTR} = enthalpy of feed water, Btu/lb

The amount of steam available for export is given by

$$\dot{Q}_{EXP} = \dot{Q}_{STMD}\ \eta_{EXP}$$

where

$\dot{Q}_{EXP}$ = steam available for export, lb/h

η_{EXP} = fraction of steam available for export

C. AUTOMATIC-EXTRACTION STEAM TURBINE

The auto-extraction steam turbine engine modeled in CELCAP is of the single extraction type. The steam turbine model includes both the condensing and non-condensing types. An automatic-extraction turbine is similar to a straight condensing turbine except that it has provisions for extracting steam at one or more points. The steam pressures at these extraction points are automatically controlled so that they are maintained at a constant value that is lower than the initial pressure, but higher than the turbine exhaust pressure. These automatic controls help the turbine maintain a constant kW output and constant extraction pressure when the flow of the extracted steam is varied. In those cases when the kilowatt output varies to meet load demands, the controls help in maintaining a constant pressure and flow in the extraction line.

A simple schematic of a single automatic-extraction steam turbine is shown in Figure 3-4. One of the major advantages of using auto-extraction turbines in cogeneration systems is that, in comparison to other steam turbines, they are

very flexible. Auto-extraction turbines can supply varying demands for extracted steam and electric energy, whether these demands vary individually or there is a variation in all the demands simultaneously. This is accomplished by varying the steam flow to the condenser to maintain the variable relation between the flow of extracted steam and the demand for electric kW.

The auto-extraction steam turbine is modeled in CELCAP by describing the performance characteristic of the steam turbine in terms of input and output parameters. These parameters include steam flow rates, engine output, inlet and extraction steam pressures and temperatures, and engine factors. The input data should have enough information so that the performance map for the given auto-extraction steam turbine can be constructed. In Figure 3-5, a typical performance map for an auto-extraction steam turbine is shown.

A simple schematic of a single automatic-extraction steam turbine/generator system is shown in Figure 3-6. The schematic shows the different locations of the steam turbine where the steam enters, exhausts, and is extracted. Steam is generated in a high-pressure boiler and passes through a throttle valve before it enters the steam turbine. For analysis, the extraction turbine generator can be assumed to be made up of two stages: a high-pressure stage between the inlet and the extraction point, and a low pressure stage between the extraction point and the exhaust point.

The governing equation for the performance of the extraction turbine/generator system can be written, based on the first law of thermodynamics, as

$$3413\ \dot{E}/\eta_{GEN} = \dot{M}_{HR}\ (h_{THR} - h_{EXT}) + \dot{M}_{EXH}\ (h_{EXT} - h_{EXH}) \tag{16}$$

$$\dot{M}_{EXH} = \dot{M}_{THR} - \dot{M}_{EXT} \tag{17}$$

where

$\dot{M}_{THR}$ = flow rate of steam entering the turbine, lb/h

$\dot{M}_{EXT}$ = flow rate of steam extracted from the turbine, lb/h

$\dot{M}_{EXH}$ = flow rate of steam exhausted from the turbine, lb/h

$\dot{E}$ = electrical power output, kW

h_{THR} = enthalpy of entering steam at inlet condition, Btu/lb

h_{EXT} = enthalpy of steam after actual expansion to extraction pressure, Btu/lb

h_{EXH} = enthalpy of steam after actual expansion to exhaust pressure, Btu/lb

The enthalpy differences $(h_{THR} - h_{EXT})$ and $(h_{EXT} - h_{EXH})$ are actual differences as opposed to ideal isentropic drops through the steam turbine. The ratio of actual enthalpy drop to ideal drop is the efficiency of the turbine.

In Figure 3-6 the expansion process in an extraction turbine is shown on an enthalpy-entropy plot. Both the actual expansion and the isentropic expansion are shown on the chart. Assuming the single extraction steam turbine to be comprised of two stages with extraction point separating the stages, the turbine stage efficiencies can be written as

$$\eta_{T1} = (h_{THR} - h_{EXT})/(h_{THR} - h'_{EXT}) \qquad (18)$$

$$\eta_{T2} = (h_{EXT} - h_{EXH})/(h_{EXT} - h'_{EXT}) \qquad (19)$$

where

η_{T1} = stage efficiency between throttle and extraction point

η_{T2} = stage efficiency between the extraction point and the exhaust

h'_{EXT} = enthalpy of steam after isentropic expansion to extraction pressure, Btu/lb

h'_{EXH} = enthalpy of steam after isentropic expansion to exhaust pressure, Btu/lb

Making an assumption that the stage efficiencies, η_{T1} and η_{T2}, are equal to the turbine efficiency, η_{TURB}, the governing equation

$$3413\ \dot{E}/\eta_g = \eta_{T1}\dot{M}_{THR}\ (h_{THR} - h'_{EXT}) + \eta_{T2}\dot{M}_{EXH}\ (h'_{EXT}-h'_{EXH}) \qquad (20)$$

can be written as

$$3413\ \dot{E}/\eta_g = [\dot{M}_{THR}\ (h_{THR}\ h'_{EXH}) - \dot{M}_{EXT}\ (h'_{EXT} - h'_{EXH})]\ \eta_{TBN} \qquad (21)$$

This can be simplified as

$$\dot{E}/(\eta_g\ \eta_{TBN}) = (\dot{M}_{THR} - \dot{M}_{EXH})\ (h_{THR} - h'_{EXT})/3413 + \dot{M}_{EXH}\ (h_{THR}\ h'_{EXH})/3413 \qquad (22)$$

This governing equation can be used to construct the performance curves of an extraction turbine with reasonable accuracy, provided that all the inputs are given.

The governing equation can also be written in terms of steam rates

$$\dot{E}/\eta_E = \dot{M}_{EXT}/TSR2 + \dot{M}_{EXH}/TSR1 \qquad (23)$$

where

$TSR1 = 3413/(h_{THR} - h'_{EXH})$, theoretical steam rate from throttle to exhaust, Btu/kWh

$TSR2 = 3413/(h_{THR} - h'_{EXT})$, theoretical steam rate from throttle to extraction, Btu/kWh

The value of TSR1 and TSR2 can be found from a Mollier diagram or from the Theoretical Steam Rate Tables.

Equation (23) can be rearranged as

$$\dot{M}_{THR} = (\dot{E}/\eta_E)TSR1 + \dot{M}_{EXT}(1 - TSR1/TSR2) \tag{24}$$

The term (1 - TSR1/TSR2) can be defined as Extraction Factor along the isentropic expansion line. Figure 3-7 shows that the total enthalpy drop, $\Delta h'$, along the isentropic line between the throttle and the exhaust points consists of two parts. The first part, $\Delta h'_2$, is between the throttle and extraction points. The second part, $\Delta h' - \Delta h'_2$ is between the extraction and the exhaust points. A pound of steam entering the turbine at the throttle point and exiting at the extraction point does $\Delta h'$ Btus of work. The loss of work due to extraction is $(\Delta h' - \Delta h'_2)$ Btu for each pound of steam extracted. To keep the load on the turbine constant, sufficient additional steam must be added to the throttle to make up this loss of $(\Delta h' - \Delta h'_2)$ Btu per pound of steam extracted. The extraction factor is the portion of a pound of steam that must be added to the throttle flow for each pound of steam extracted. The theoretical extraction factor is given by

$$\begin{aligned} \text{Theoretical Extraction Factor} &= (\Delta h' - \Delta h'_2)/\Delta h \\ &= 1 - \Delta h'_2/\Delta h' \\ &= 1 - TSR1/TSR2 \end{aligned} \tag{25}$$

The extraction factor determined from the actual process 1-2-3 in Figure 3-7 is expressed as

$$\text{Extraction Factor} = 1 - \Delta h_2/\Delta h \tag{26}$$

The Extraction Factor is normally presented as a function of the ratio of theoretical steam rates (TSR1/TSR2). Therefore, the Extraction Factor is expressed as

$$\text{Extraction Factor} = 1 - C\ (\text{TSR1/TSR2}) \qquad (27)$$

where C is an empirical correction factor, whose value depends on the type of extraction turbine, condensing or noncondensing. A plot of the extraction factor based on this empirical correction factor is shown as a function of the theoretical steam rate ratio in Figures 3-8 and 3-9 for condensing and non-condensing single-extraction steam turbines, respectively. This relationship and the assumption that all extraction charts are made up of straight lines parallel to each other and equally spaced make possible this estimating method. Such an assumption introduces an error that is more than compensated for by the simplicity it makes possible in the estimating method. The value of C for the plots shown in Figures 3-8 and 3-9 are 0.857 for condensing turbines and 0.902 for noncondensing turbines, respectively.

The term (1 - TSR1/TSR2) in Equation (24) can be substituted by the actual extraction factor [1 - C(TSR1/TSR2)] from Equation (12) to obtain

$$\dot{M}_{THR} = (E/\eta_E)\text{TSR1} + \dot{M}_{EXT}\ [1 - C(\text{TSR1/TSR2})] \qquad (28)$$

Equation (28) is used for generating the performance chart for a single-automatic-extraction unit. A typical performance chart of a single-automatic-extraction turbine is shown in Figure 3-5.

In the performance chart shown in Figure 3-5, the family of parallel lines define the throttle steam at given kW output and extraction flow. The minimum exhaust line shows the relation between the kW output produced on extracted steam alone and the corresponding throttle steam flow. This curve intersects each of the constant extraction-flow curves at the throttle flow that equal the sum of the minimum steam flow to the exhaust and the extraction steam. The exhaust sections of the turbine often pass as much exhaust flow as needed to produce the rated output at zero extraction. Additional output can be generated by admitting more steam at the throttle and extracting it. This is the maximum flow to the exhaust line. The other limits are the maximum throttle flow and maximum generator output.

The first step in developing a calculation procedure for auto-extraction steam turbines involves constructing performance curves for them. The governing equation, expressed in terms of various steam rates and engine power ratings, are used for this purpose. For a given engine, the steam conditions such as the pressure and temperature of the throttle steam, the extraction and exhaust steam pressures, and the maximum extraction flow rate are usually given. The

maximum throttle steam rate and the minimum exhaust flow are sometimes not given, and thus must be estimated. A typical performance map for an auto-extraction steam turbine is shown in Figure 3-5. The actual construction of this map requires computing points A through F. To compute these points, the following calculations are used.

1. The Full-Load Non-extraction Throttle Flow

The full-load non-extraction throttle flow, $\dot{M}_{THR,A}$, is obtained by setting the engine output E_A at the rated generator output $\dot{E}_A$ (0.8 power factor) in Equation (27)

$$\dot{M}_{THR,A} = (\dot{E}_A/\eta_{E,F})\ TSR1 \tag{29}$$

where $\eta_{E,F}$ is the full load efficiency with no extraction. For a condensing turbine, the value of $\eta_{E,F}$ for the turbine rating up to 7500 kW is given in Table 3-1. For noncondensing, $\eta_{E,F}$ is given in Table 3-2. For other sizes of units that are not listed in these tables the value of $\eta_{E,F}$ should be estimated. The value of $\eta_{E,F}$ taken from the tables are in the proper magnitude, but may be higher or lower than the actual performance guaranteed for specific turbine. In most cases, however, regardless of design, the error for efficiencies read from these tables will be less than 5%.

2. The Half-Load Non-extraction Throttle Flow

The half-load non-extraction throttle flow, $\dot{M}_{THR,B}$, (point B on the map in Figure 3-5) is obtained by setting the engine output E in Equation (28) to $\dot{E}_A/2$. Rewriting the equation with this substitution

$$\begin{aligned}\dot{M}_{THR,B} &= [(\dot{E}_A/2)/\eta_{E,H}]\ TSR1 \\ &= (\dot{E}_A/\eta_{E,F})TSR1\ [\eta_{E,F}/(2 * \eta_{E,H})]\end{aligned} \tag{30}$$

where $\eta_{E,H}$ is the engine efficiency at half-load with no extraction.

Substituting Equation (14) for $\dot{M}_{THR,A}$ and defining $\eta_{E,F}/(2 * \eta_{E,H})$ as the half-load flow factor, H, Equation (15) becomes

$$\dot{M}_{THR,B} = \dot{M}_{THR,A} * H \tag{31}$$

The half-load flow factors, H, presented in Table 3-1 for condensing turbines and Table 3-2 for noncondensing turbines are approximations that assume that the throttle flow versus output curve at no extraction will be a straight line. These tables assume that all turbines of the same rating, regardless of design, will have the same half-load flow to full-load flow relationship. Obviously this relationship is not a constant one, but the error introduced by this assumption is negligible.

3. The Full-Load Throttle Flow

The full load throttle flow, $\dot{M}_{THR,C}$ (Point C in Figure 3-5), is obtained by combining Equations (13) and (14) at the maximum extraction flow limit. With this substitution, $M_{THR,C}$ is given by

$$\dot{M}_{THR,C} = \dot{M}_{THR,C} + \dot{M}_{EXT,C} \left[1 - C(TSR1/TSR2)\right] \qquad (32)$$

where C is the correction factor that is 0.857 for condensing turbines and 0.902 for noncondensing turbines.

If the maximum throttle flow is not given by the user as the input information, then the calculated $M_{THR,C}$ will be designated as the maximum throttle flow through Point C. When a higher value of maximum throttle flow $M_{THR,C}$ is given, the power output at point C' can be calculated by using Equation (32).

$$\dot{E}_{C'} = (\dot{E}_A/\dot{M}_{THR,C'}) \quad \dot{M}_{THR,C'} - \dot{M}_{EXT,C} \left[1 - C(TSR1/TSR2)\right] \qquad (33)$$

The lines for extraction flows less than maximum can be drawn parallel to line AB in Figure 3-5 at distances proportional to the extraction quantity. Thus, the distance between points A and C, which represents maximum extraction flow, is divided into several equal parts and lines are drawn parallel to AB passing through these points as shown in Figure 3-5.

4. The Minimum Flow to Exhaust Limits

If the value of minimum flow to exhaust, $\dot{M}_S$, is not available to the user, Figure 3-10 can be used to select an approximate value for units of sizes between 500-kW and 7500-kW rated output. A minimum amount of flow to the exhaust is necessary to prevent overheating the low-pressure section. Points are then plotted on the extraction lines where the throttle flow equals extraction flow, plus minimum flow to the exhaust ($\dot{M}_{THR} = \dot{M}_{EXT} + \dot{M}_S$). A straight line through such points forms the left boundary of the diagram (Figure 3-5).

5. The Maximum Flow to Exhaust Limits

The line of maximum flow to exhaust, which is shown in Figure 3-5, is obtained by joining the points on each extraction line where throttle flow equals nonextraction rated load throttle flow, $M_{THR,A}$ at point A, plus the extracted flow, i.e., $\dot{M}_{THR} = \dot{M}_{EXT} + \dot{M}_{THR,A}$.

In this estimating method, an assumption is made that all turbines are designed so that their exhaust sections are large enough to enable the turbine to carry full-rated output with the extraction pressure held constant and no extraction taken from the turbine.

6. The Maximum Generator Output at 1.0 Power Factor

The usual turbine-generator set has an 0.80 power factor generator, and a turbine carrying a full kVA on the generator at 1.0 power factor. This is indicated as the maximum generator output at 1.0 power factor on Figure 3-5.

D. BACK-PRESSURE STEAM TURBINE

This is a noncondensing steam turbine and is commonly referred to as a back-pressure steam turbine. It takes steam at the boiler pressure and temperature and exhausts it at atmospheric pressure or above. For equal power outputs, back-pressure turbines may require two to five times the steam flow required by the condensing turbine. When the back-pressure turbines are used in industrial plants, heat energy in the exhaust steam is used for heating, drying, cooking, and various other process uses. The thermal efficiency of the system when the exhaust steam is used in process may be as high as 70% to 75%. However, the thermal efficiency can be as low as 10% if the heat energy in the exhaust steam is not utilized. The back-pressure steam turbines are widely used in cogeneration systems. Approximately 30% of the steam turbines sold for power generation in industrial plants are of the back-pressure type.

The performance characteristics of a typical back-pressure steam turbine are shown in Figure 3-11. The performance data are based on an actual General Electric turbine with a rating of 2500 kW. There are two important points to be noted in Figure 3-11. The first point is that the generated power is directly proportional to the steam passing through the turbine. The second point is that the heat rate (Btu/kWh) is inversely proportional to the generated power.

The model of the back-pressure steam turbine used in CELCAP uses as input the data on the turbine performance. The turbine model estimates the fuel flow rate, the throttle steam flow rate, and the amount of steam exported. At the rated output of the turbine, E_{ED}, the export steam and the fuel consumption rates are given by

$$\dot{M}_{THR} = \dot{E}_{ED} * \dot{S}_{FD}$$

$$\dot{M}_{EXP} = \dot{M}_{THR} * \dot{E}_{XLIM}$$

$$\dot{M}_{FUEL} = \dot{M}_{THR} * (h_{THR} - h_{EXH})/(\eta_B * \eta_{COMB})$$

where

$\dot{E}_{ED}$ = rated turbine output, kW

$\dot{S}_{FD}$ = steam rate at $\dot{E}_{ED}$, lb/kWh

$\dot{M}_{THR}$ = maximum throttle steam rate, lb/h

$\dot{E}_{XLIM}$ = percentage of exhaust steam to be used as export steam (decimal)

$\dot{M}_{EXP}$ = export steam rate, lb/h

$\dot{M}_{FUEL}$ = fuel flow rate to boiler at the rated turbine output, Btu/h

h_{THR} = enthalpy of throttle steam, Btu/lb

h_{EXH} = enthalpy of exhaust steam, Btu/lb

η_B = boiler efficiency

η_{COMB} = combustion efficiency (assumed to be 0.98 in this model)

When the turbine is operating at a fraction of its rated capacity, the throttle steam flow rate and the fuel flow rate are given by

$$\dot{M}_{FRC} = \dot{S}_{FRC} * \dot{E}_{FRC}$$

$$\dot{S}_{FRC} = \dot{S}_{FD} + (\dot{E}_{ED} - \dot{E}_{FRC}) * W_{CD}$$

$$W_{CD} = (\dot{S}_{FP} - \dot{S}_{FD})/(\dot{E}_{ED} - \dot{E}_{EP})$$

$$\dot{M}_{FUELF} = \dot{M}_{FRC} * (h_{THR} - h_{EXH})/(\eta_B * \eta_{COMB})$$

where

$\dot{E}_{FRC}$ = fractional power output of the turbine, kW

$\dot{M}_{FRC}$ = throttle steam rate at the turbine part load, $\dot{E}_{FRC}$, lb/h

$\dot{S}_{FRC}$ = steam rate at fractional power output, $\dot{E}_{FRC}$, lb/kWh

$\dot{M}_{FUELF}$ = fuel flow rate at the turbine fractional power output, $\dot{E}_{FRC}$, Btu/h

$\dot{E}_{EP}$ = turbine power output at part load, kW

$\dot{S}_{FP}$ = steam rate at part load, $\dot{E}_{EP}$, lb/kWh

W_{CD} = part-load steam rate factor for given part-load data, lb/h-kW

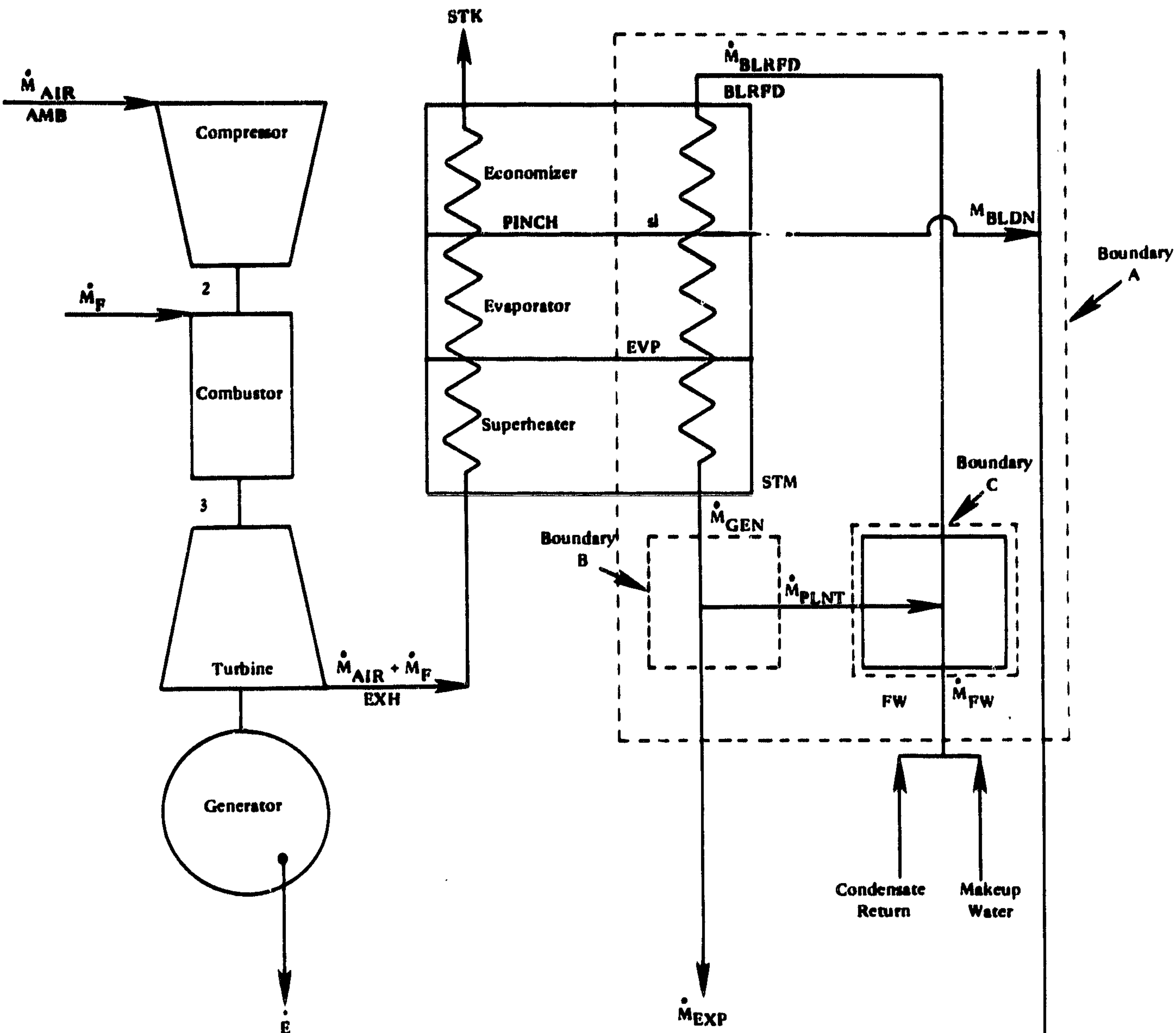

Figure 3-1. Schematic of Combustion Turbine/Exhaust Heat Boiler Cogeneration Plant

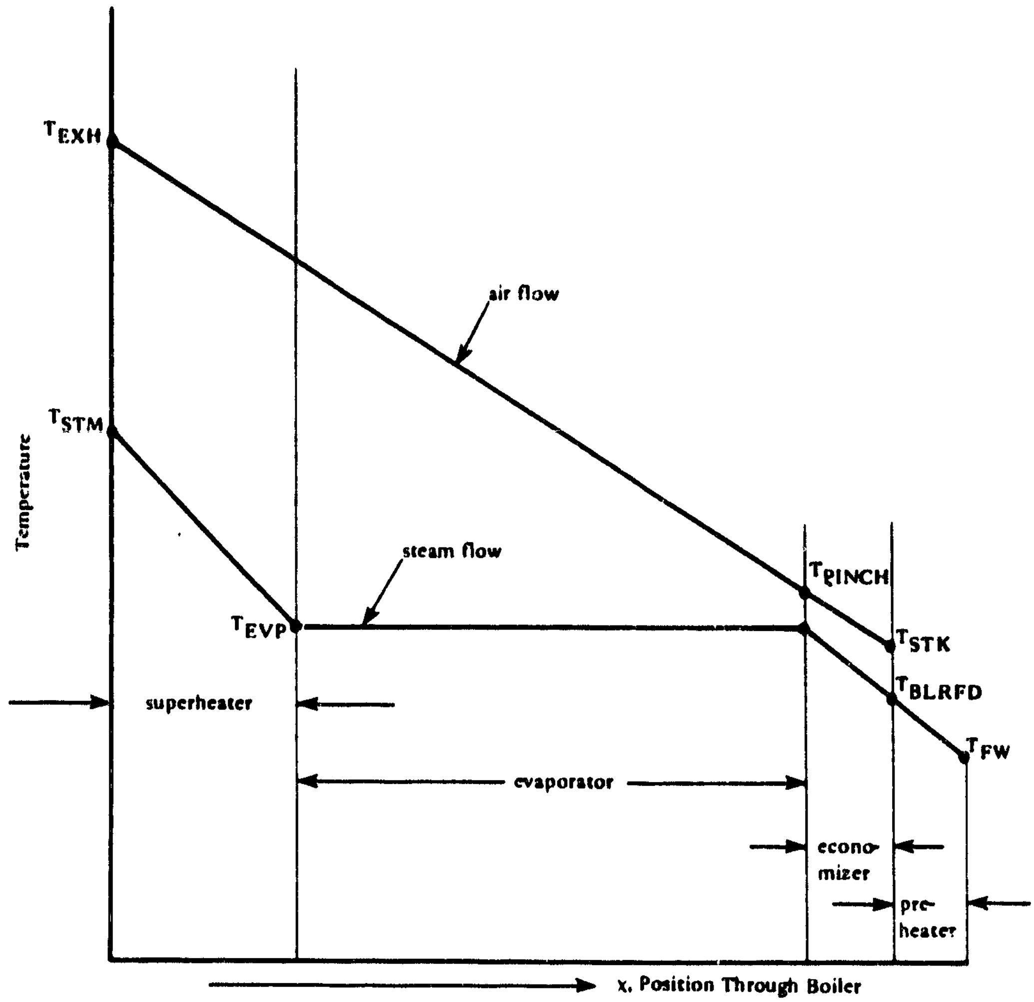

Figure 3-2. Temperature Profiles in Exhaust Heat Recovery Boiler

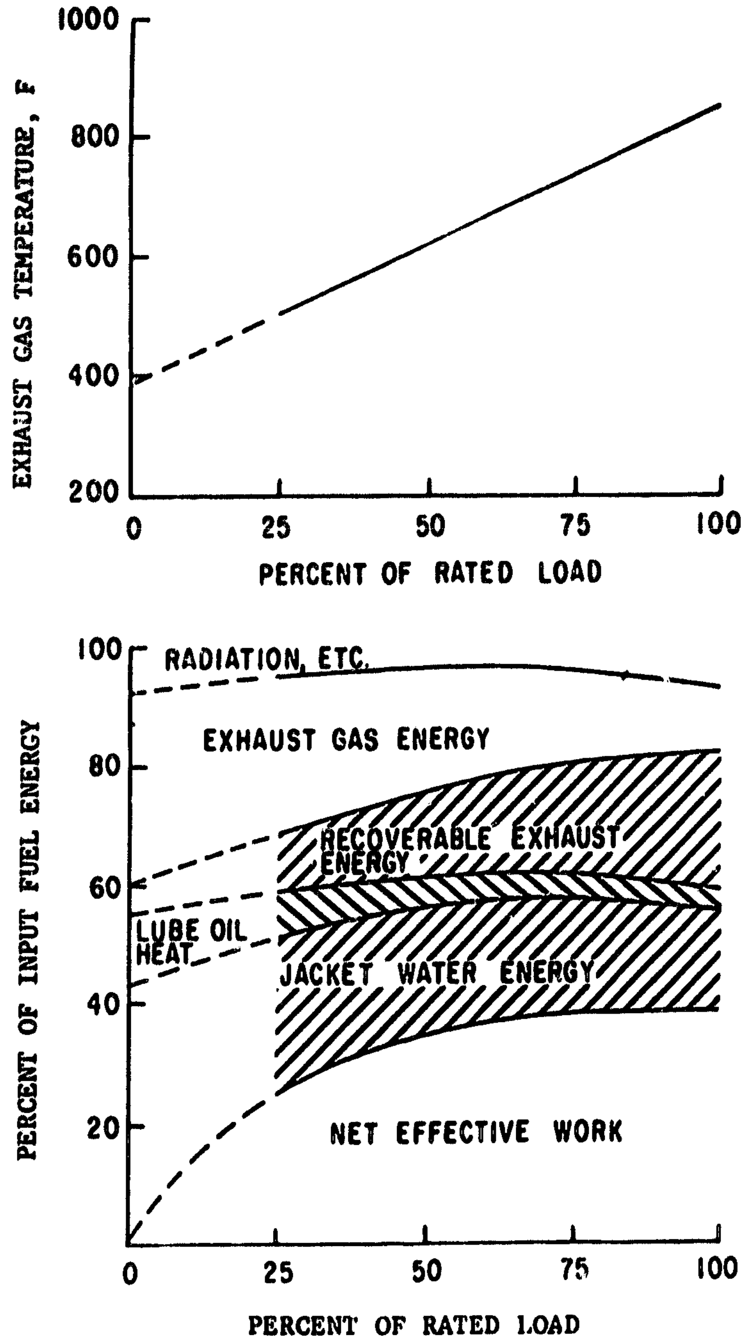

Figure 3-3. Estimated Exhaust Gas Temperature and Part-Load Heat Balance of a Typical Turbo-Charged Diesel Engine

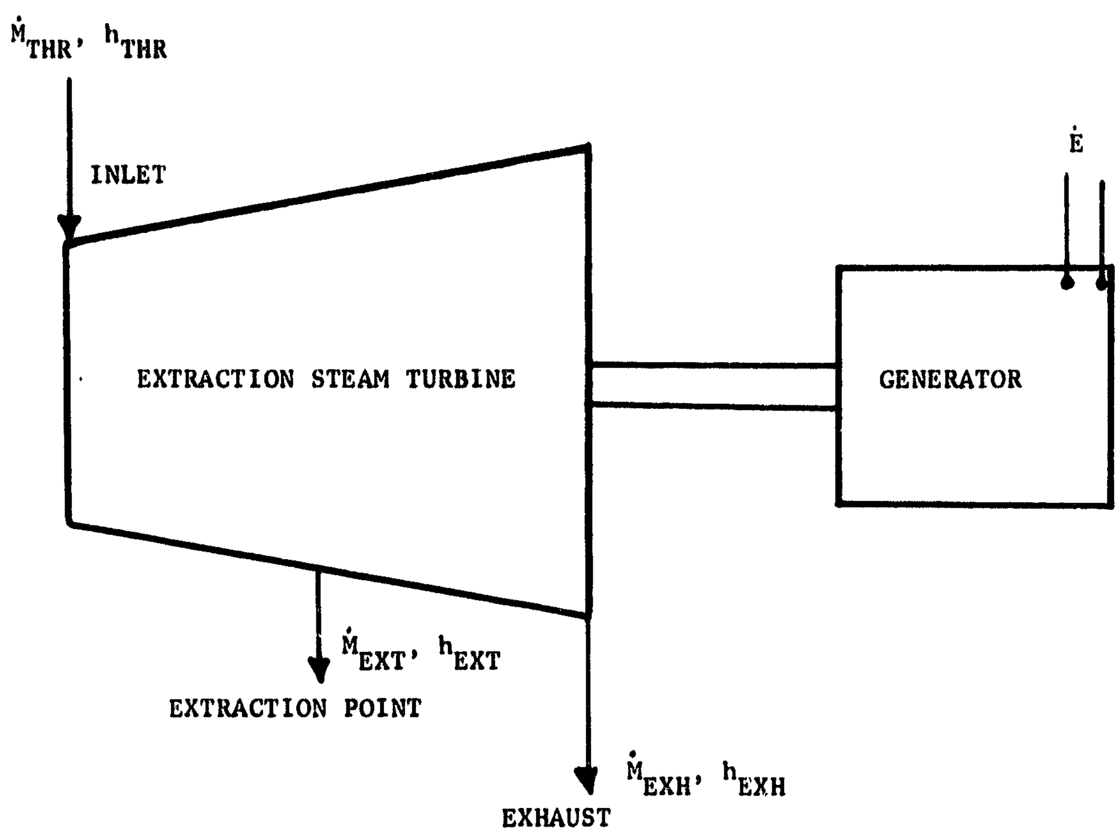

Figure 3-4. Single-Automatic Extraction Steam Turbine

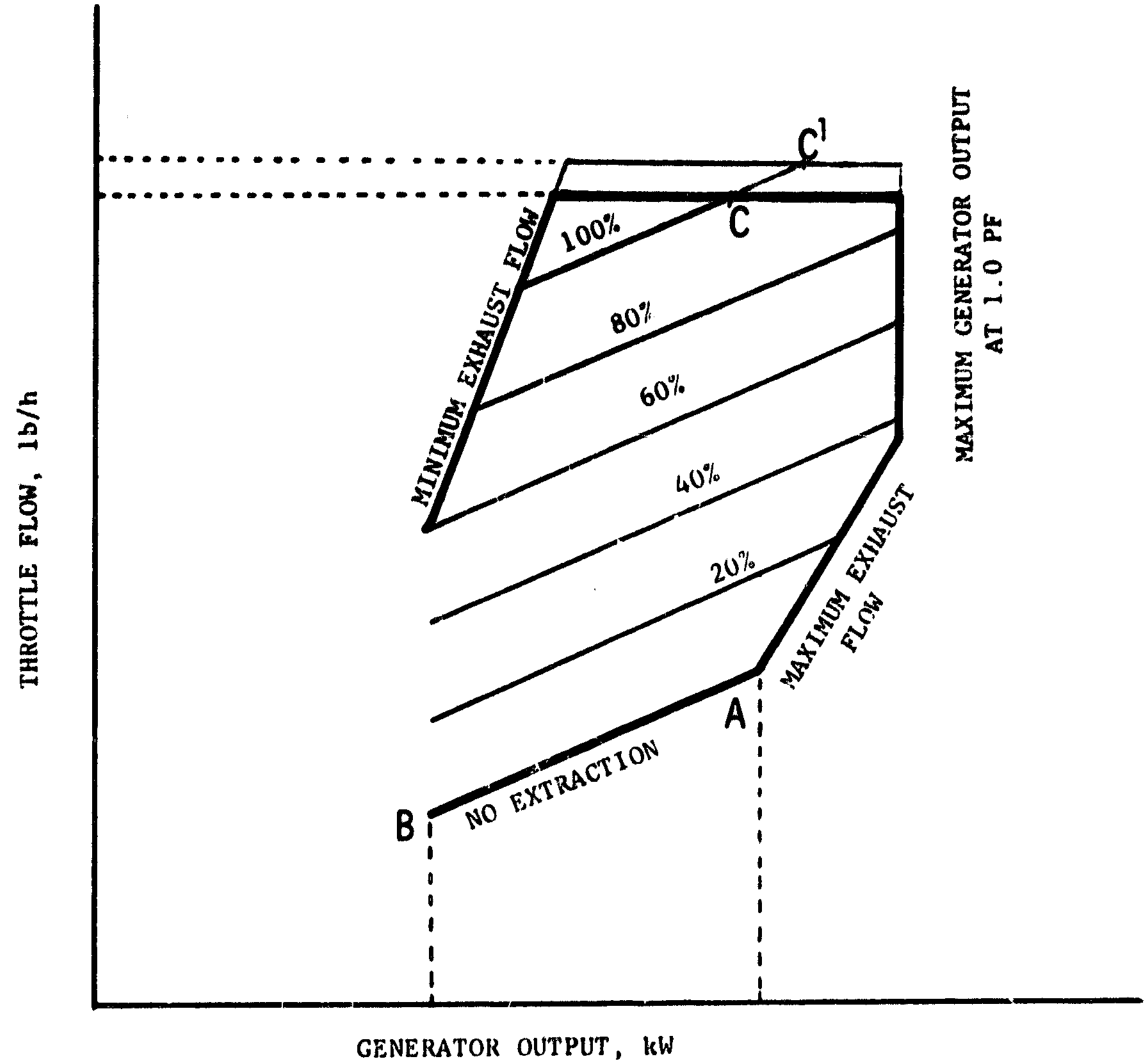

Figure 3-5. Typical Performance Map of a Single-Automatic Extraction Steam Turbine

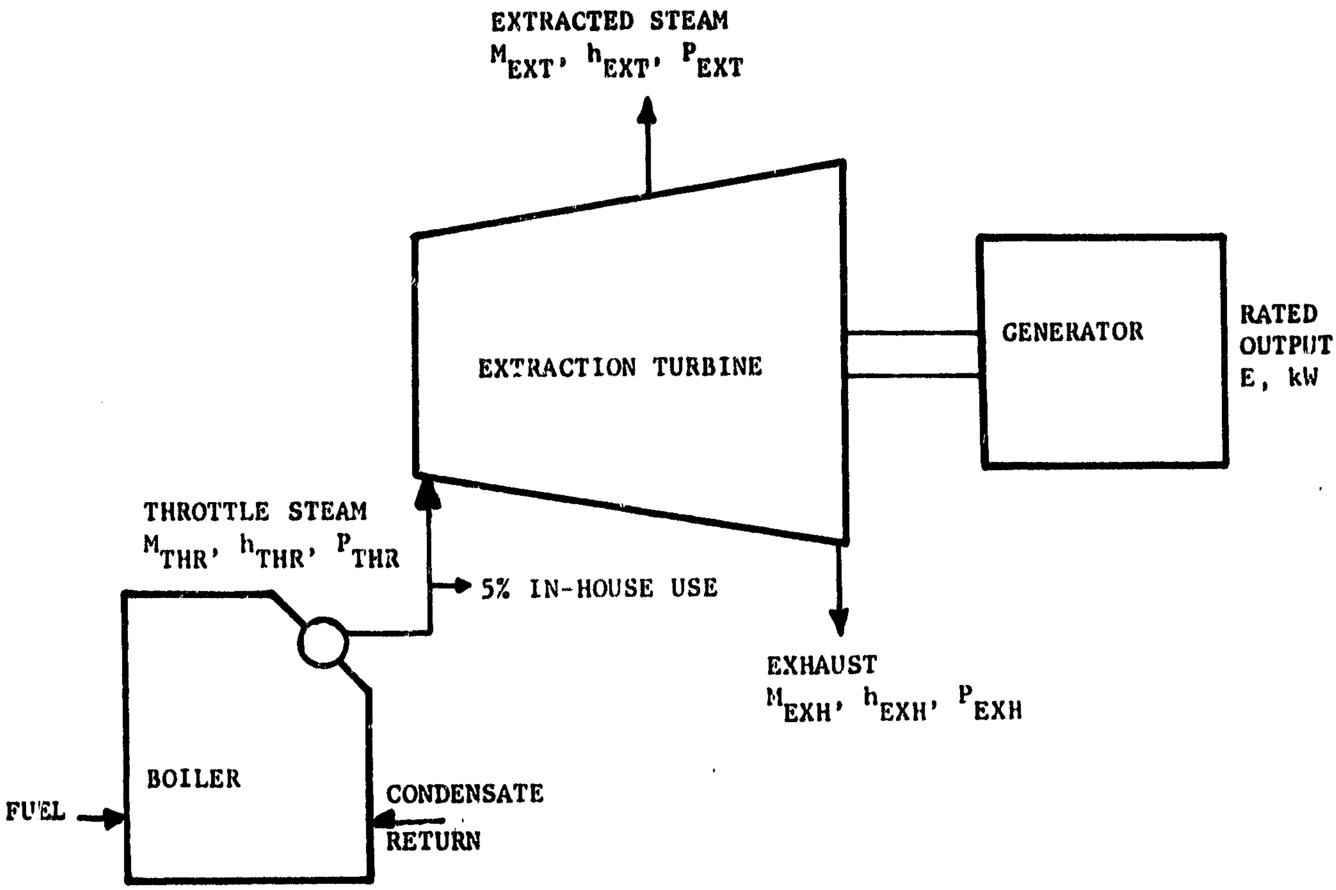

Figure 3-6. Schematic of a Single-Automatic Extraction Steam Turbine/Generator

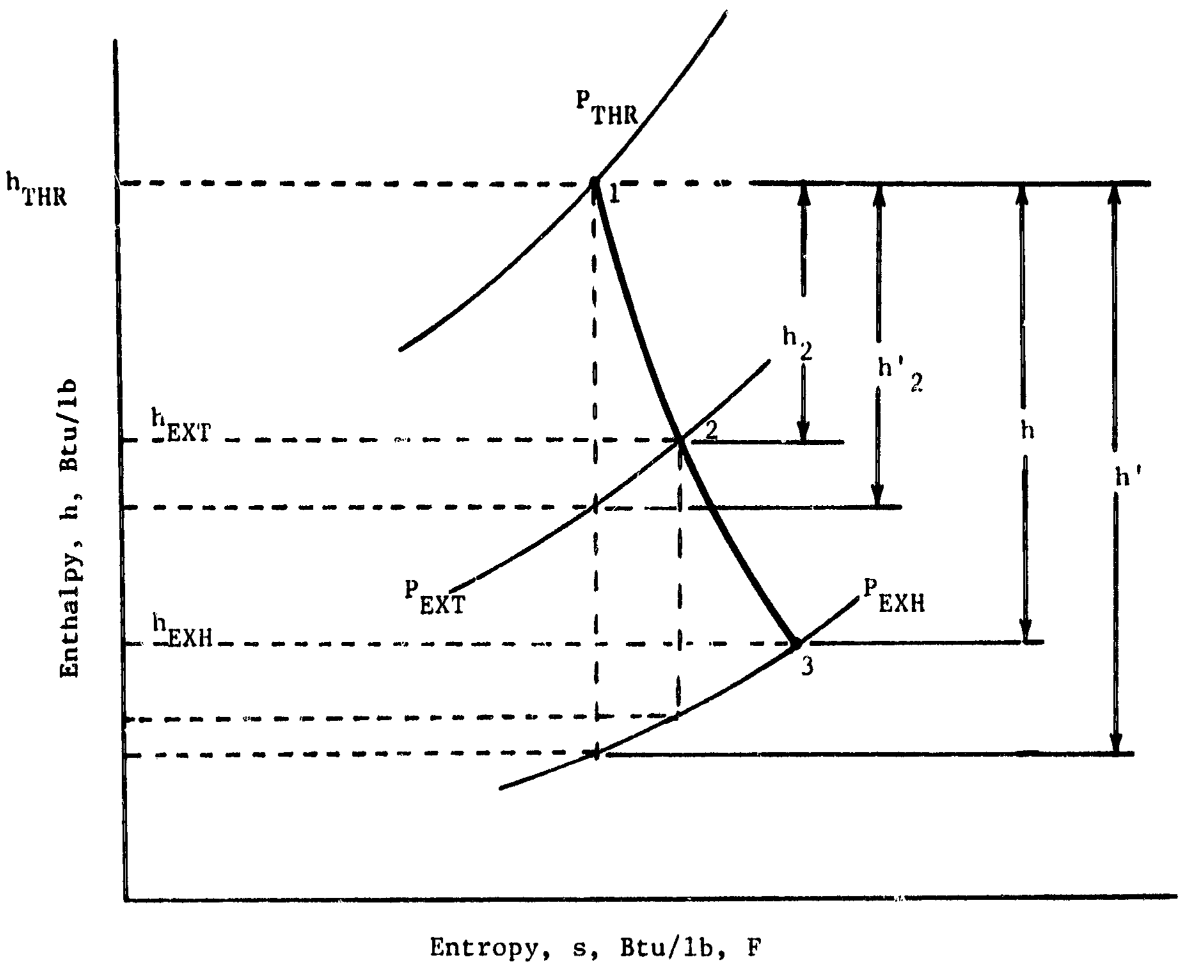

Figure 3-7. Expansion Process for a Single-Extraction Steam Turbine on h-s Chart

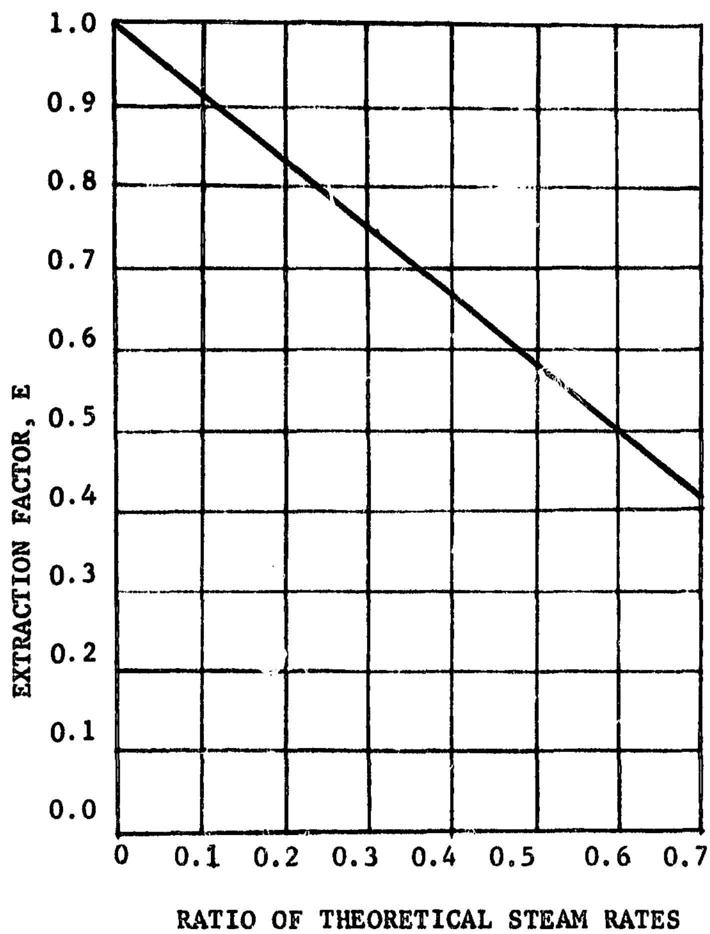

Figure 3-8. Extraction Factor Versus Ratio of Theoretical Steam Rates of Condensing Single-Automatic Extraction Steam Turbine

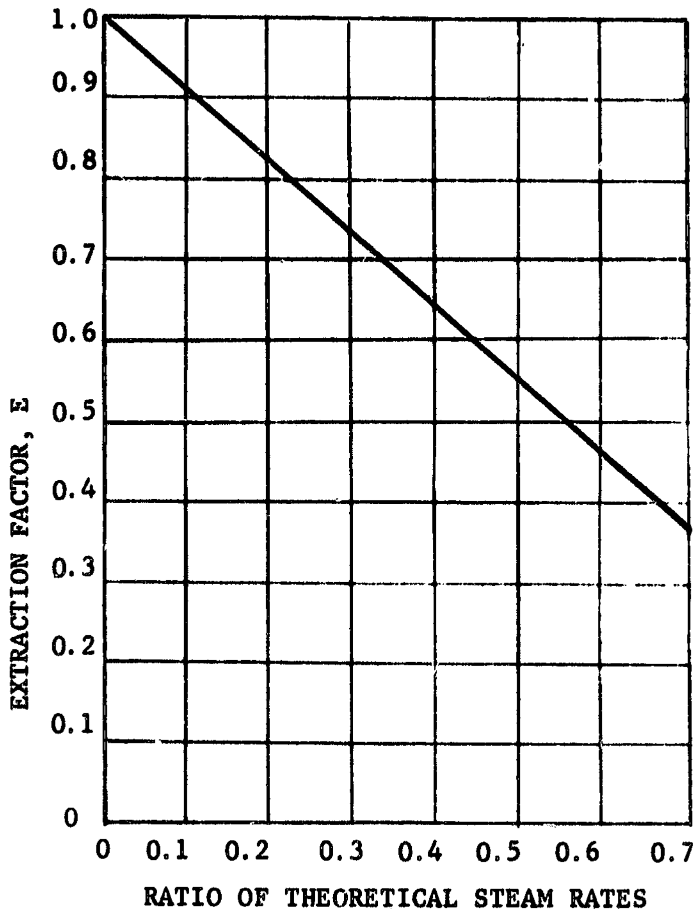

Figure 3-9. Extraction Factor vs Ratio of Theoretical Steam Rates of Noncondensing Single-Extraction Steam Turbines

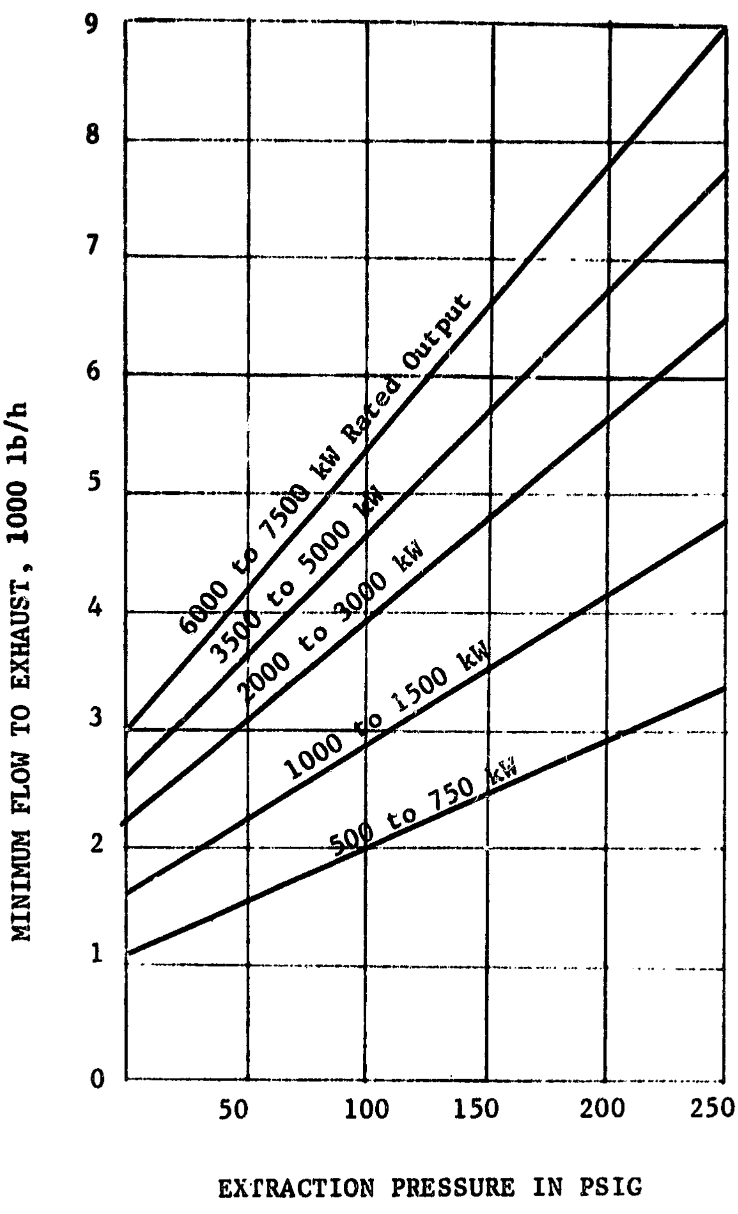

Figure 3-10. Plot of Minimum Flow to Exhaust Versus Extraction Pressure

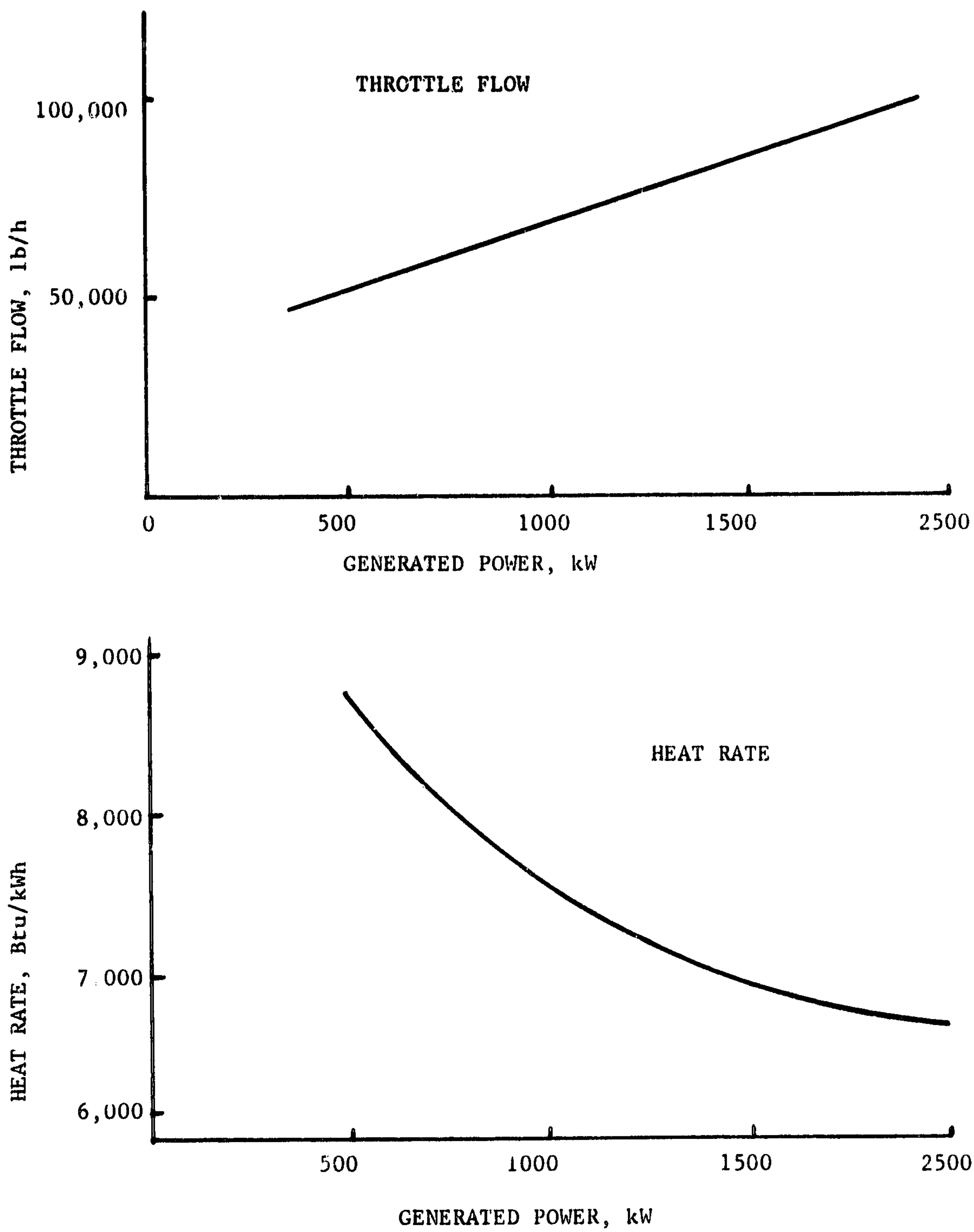

Figure 3-11. Performance of a Typical Back-Pressure Steam Turbine

Table 3-1. Full-Load Nonextraction Efficiencies for Condensing Single-Automatic Extraction Steam Turbines

Rating, kW 0.8 pf	Main Pressure, psig						
	150	200	250	300	400	600	650
	Efficiency						
500	0.600	0.595	0.585	0.580	0.565	0.545	
625	0.615	0.610	0.605	0 600	0.580	0.560	
750	0.630	0.625	0.620	0.610	0.595	0.575	
1000	0.650	0.645	0.640	0.630	0.620	0.600	
1250	0.665	0.660	0.650	0.645	0.635	0.615	
1500	0.675	0.670	0.665	0.660	0.645	0.630	
2000	0.690	0.685	0.680	0.675	0.665	0.645	
2500	0.700	0.695	0.690	0.685	0.675	0.660	
3000	0.710	0.705	0.700	0.695	0.685	0.670	
3500	0.715	0.710	0.705	0.700	0.690	0.680	
4000	0.710	0.715	0.710	0.705	0.700	0.685	
5000	0.725	0.720	0.715	0.710	0.705	0.695	0.685
6000	0.735	0.730	0.725	0.710	0.715	0.705	0.695
7500	0.740	0.735	0.730	0.725	0.720	0.715	0.705

Table 3-2. Full-Load Non-extraction Efficiencies for Noncondensing Single-Automatic Extraction Steam Turbine

Rating, kW 0.8 pf	Main Pressure, psig						
	150	200	250	300	400	600	650
	Efficiency						
2000	0.700	0.690	0.685	0.675	0.660	0.630	----
2500	0.710	0.705	0.695	0.690	0.675	0.645	0.620
3000	0.720	0.715	0.705	0.700	0.690	0.660	0.635
3500	0.725	0.720	0.715	0.710	0.695	0.670	0.650
4000	0.730	0.725	0.720	0.715	0.695	0.680	0.660
5000	0.735	0.735	0.730	0.725	0.715	0.695	0.675
6000	0.740	0.735	0.735	0.730	0.725	0.705	0.685
7500	0.745	0.740	0.740	0.735	0.730	0.715	0.700

Table 3-3. Half-Load Factor for Condensing Single-Automatic Extraction Steam Turbine

Rating, kW at 0.80 pf	Factor, H
500	0.590
625	
750	
1000	0.585
1250	
1500	
2000	0.580
2500	
3000	
3500	0.575
4000	
5000	
6000	0.570
7000	

Table 3-4. Half-Load Factor for Noncondensing Single-Automatic Extraction Steam Turbine

Rating, kW at 0.80 pf	Factor, H
2000	
2500	0.630
3000	
3500	
4000	0.625
5000	
6000	
7000	0.620

SECTION IV

LOADS AND ECONOMIC MODELS

A. ELECTRICAL AND STEAM LOADS MODEL

The primary function of a cogeneration system is to supply electricity and thermal energy to the user. The energy supplied by the system is used for meeting the needs of the application of the user. Depending on the user, the application may vary, from an institutional user with electric lighting and hot water/space heating application to an industrial user with electric furnaces and thermal processes needing electricity and high temperature steam. However, as far as the cogeneration system is concerned, all these applications represent the electric and steam demand that should be supplied by the system. The pressure and temperature of the steam needed for each application may be different, depending on the user. In effect, the user is represented to the cogeneration system as electric and steam load profiles with a specific value for the steam pressure. In the computation of the performance of the cogeneration system, hourly load profiles are commonly used.

A description of how to develop the load profiles of the user is given in the section on methodology. That section describes how two load profiles can be used for each month of the year to represent the load demand of the user for the whole year. The two load profiles for each month are the working day profile and non-working day profile. It is explained in Section II that the weather variation during each month is not very significant and one average profile can be used to accurately represent all the days of the month. It is also explained in Section II that because of the large difference in energy usage between a working day and a non-working day, two separate average profiles representing working and non-working days should be used. A total of 24 hourly profiles are used to represent the electric demand of the user for the whole year. Similarly, another 24 profiles are used to represent the steam demand of the user for the whole year.

The manner in which the load profiles are read in the CELCAP code is shown in Figure 4-1. Each hourly load profile is made up of 24 numbers representing the demand at each hour of the day. A total of 24 profiles is used for electric demand and another 24 for steam demand of the user. The cogeneration system performance is computed on an hourly basis for a working day and a non-working day for each month of the year. The results from this computation provide the hourly performance and also the performance of the system on a daily basis. The performance of the cogeneration system is computed using the daily totals; the performance of the system on a monthly basis is computed by adding two products. The first one is the product of number of the working days in the month and the working day totals. The second is the product of number of the non-working days and the non-working day totals. Similarly, the annual performance of the cogeneration system is computed by adding all the monthly totals for the year.

B. ECONOMIC MODEL

The function of the economic model in the cogeneration system model is to calculate the various economic parameters needed for evaluating the cogeneration system. The economic model used in the CELCAP code is based on the economic evaluation procedure outlined in Section II. The model uses the input information on the economic parameters and calculates the monthly and annual cost of operating the cogeneration system. In Figure 4-2, an overall block diagram of the economic model is given. In the CELCAP code, the amount of purchased electricity and fuel used for supplying the electric and thermal needs of the user were first calculated. The monthly and annual costs of providing this energy to the user are then calculated using the input data on fuel and purchased electric prices and other economic data.

The input data needed for computing the various economic parameters of the cogeneration system are the following: (1) prices of various fuels used by the prime movers in the system, (2) operation and maintenance costs of prime movers, (3) purchased electricity rates, (4) escalation rates of fuel and purchased electric costs and rates, and (5) discount rate. In the CELCAP code, the heat engine and the auxiliary boiler models calculate the amount of fuel needed to generate electricity and steam to meet the user's demand. Also calculated in the code is the amount of electricity purchased to meet the user's electric demand that is not provided by the on-site power. The monthly totals of the amounts of fuel and purchased electricity are used along with the input economic data to calculate the monthly and annual costs. Also calculated with these costs are the life-cycle operating costs of the prime movers, boiler, the purchased electricity, and the total system. The life-cycle period and the escalation rates provided in the input data are used for this purpose.

The economic model calculates the monthly, annual, and life-cycle operating cost of running the cogeneration system to meet the user's electric and steam demands. Several criteria are used for judging whether the cogeneration system under consideration is cost effective or not. The first is the discounted SIR. This is the ratio of savings in life-cycle operating cost due to the cogeneration system under consideration compared with the existing system on-site to the capital cost of installing the new system. For the candidate system to be economically better than the existing system, the SIR should be greater than one. The second criteria for judging the economic viability of the system is the simple payback period. It is the ratio of savings in annual cost due to operating the cogeneration system to the capital cost of installing the system. Many institutional and commercial users have a maximum limit on this payback. The calculated payback should be below the limit set by the users. The third criteria is whether there are any alternative cogeneration options that have a higher SIR and lower payback period.

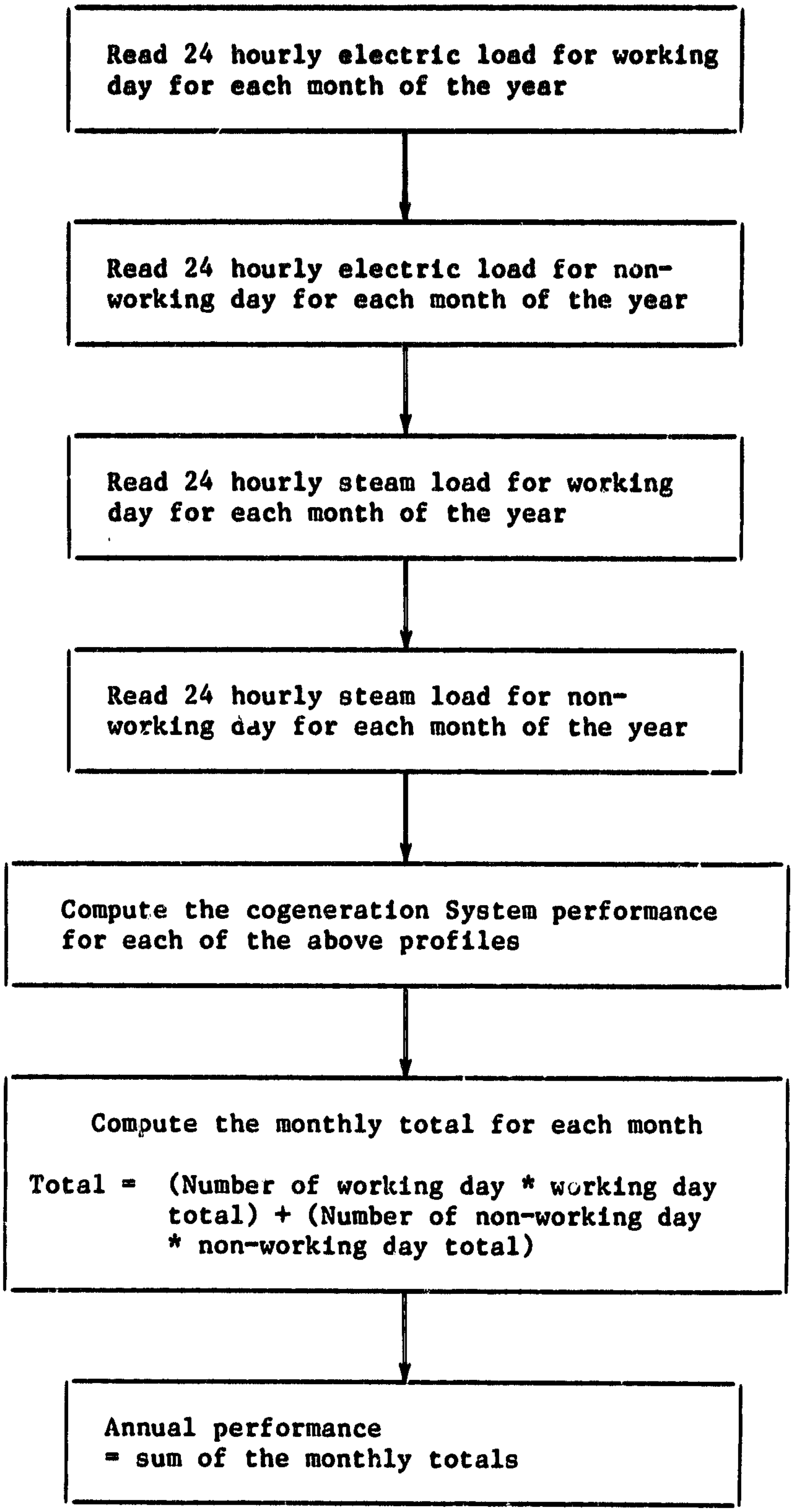

Figure 4-1. Block Diagram of the Load Model

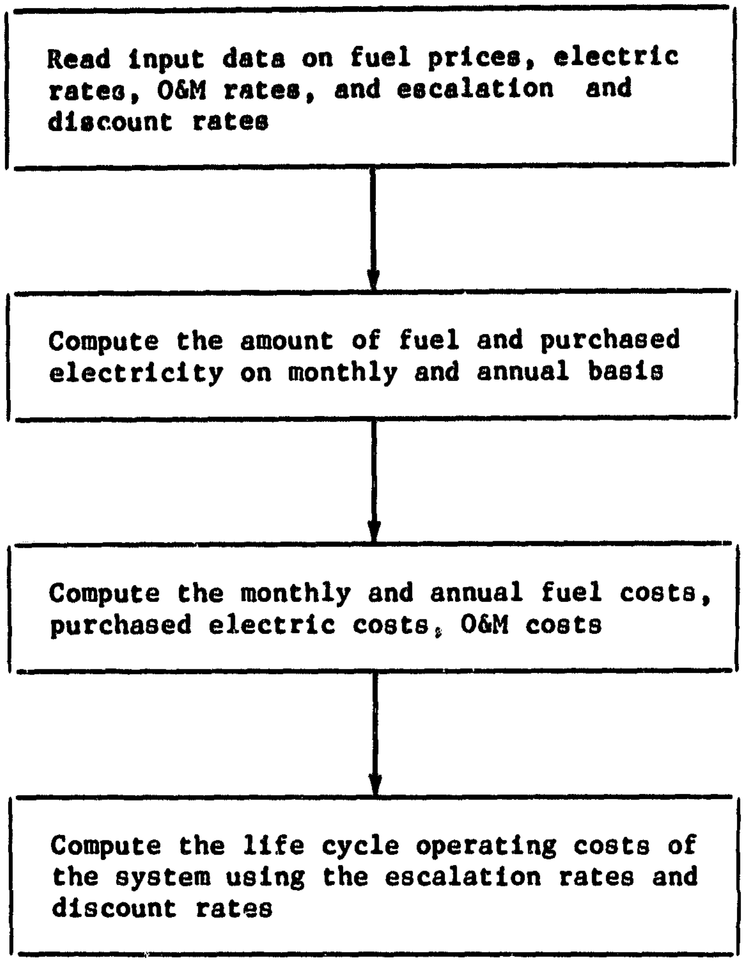

Figure 4-2. Overall Block Diagram of the Economic Model

SECTION V

MODEL STRUCTURE AND FUNCTIONS

The CELCAP computer model is organized to calculate the performance of the candidate cogeneration system. Using the input information on the candidate cogeneration system and the user's energy demand, the model calculates the performance of the candidate system. The output results from the model are the energy performance numbers and the annual and life-cycle operating cost numbers. These performance data are used along with the capital cost data of the system in the evaluation of the candidate cogeneration system.

A. BLOCK DIAGRAM OF THE CELCAP MODEL

The CELCAP computer model can be subdivided into four phases of operation. An overall block diagram of the CELCAP model is shown in Figure 5-1. In the first phase, all the input data on the candidate cogeneration system and the user's electrical and steam demands is read. In the second phase, all the input data are processed and a performance analysis of the engines and boiler is made. Also carried out in this phase are the load analysis and the comparison of the electric and thermal loads with the engine outputs. In the third phase of operation, the model calculates the performance of the cogeneration system for meeting the user's electrical and steam loads. In the fourth phase of the operation, the model calculates the annual and life cycle operating costs of the system. For this calculation, the performance results from the third phase and the input data on the energy escalation rates are used.

The input data provided to the CELCAP code can be divided into three parts: (1) the data on the candidate cogeneration system, (2) the electric and steam demand data of the user, and (3) data on the fuel and purchased electricity costs, and the O&M costs of the engines along with the escalation rates for all these costs. A block diagram of the input data read in the CELCAP code is shown in Figure 5-2. The input data on the candidate cogeneration system include the information on the control mode, boiler feed water and steam temperatures, number and type of engines, and the design and part load performance of the engines. The second part of the input data consists of the electrical and steam demand of the user. Hourly demand profiles of working day and non-working day for each month of the year are read. Also input here are the peak and off-peak hours of working and non-working days of summer and winter months. The last part of the input data includes the information on fuel and purchased electricity prices, fixed and variable O&M costs, maximum peak and off-peak purchased electric demand, duration of plant construction and life of the plant, short and long-term escalation rates for fuel, electricity, and O&M costs, and the discount rates.

The CELCAP model reads all the input data and processes the data into a form that can be used in the performance calculation. A block diagram of the various steps in this part of the model is shown in Figure 5-3. The model calculates the maximum electrical and steam outputs from each engine at the ambient conditions. For the gas turbine, the electrical output is calculated

at the design and ambient conditions. For the auto-extraction steam turbine, all the points needed to draw the performance map of the turbine are calculated. After the limiting electrical and steam outputs of the engines are calculated, the total electrical and steam outputs from all the engines in the candidate system are computed. These totals are compared with the actual electric and steam demands of the user. The candidate system operates to meet the demand according to the control mode chosen.

Three control modes are built into the CELCAP model so that the cogeneration system can operate in any one of them. In the first control mode, all the engines in the system operate at their full rated capacity. The total electrical output is compared to the demand of the user. If the demand is lower than the generated output, the excess electricity is sold to the utility. If the demand is higher than the generated electricity, the shortfall in electricity is made up by purchasing it from the utility. A block diagram of the operation of the system in this control mode is shown in Figure 5-4. As far as the steam is concerned, if the demand is lower than that generated, the excess steam is discarded. Any shortfall in the steam output is made up by the auxiliary boiler. The auto-extraction steam turbine has a degree of flexibility in controlling the amount of steam extracted at its rated electrical output. This capability of the turbine is used, as much as possible, to match the steam output and the demand.

In the second control mode, the engines follow the electrical load up to their rated capacity. A block diagram of the performance calculation for this mode is shown in Figure 5-5. If the electrical output is greater than the electrical load, the peak engines are turned off. If the output is still larger than the load, the engines are run at a fractional load factor equal to the ratio of the load to the total rated cogeneration capacity. If the load is larger than the total cogeneration capacity, but smaller than the sum of cogeneration and peak capacity, then the fractional load factor is the ratio of the load to the sum of cogeneration plus the peak engine capacities. The model calculates the total amount of steam generated by the cogeneration system at this point and compares it to the steam load. The remaining calculations are the same as those for the first control mode explained earlier.

In the third control mode of operation, the engines are run in such a way that they follow the steam load up to their rated capacity. A block diagram of the operation of the various engines and boilers is shown in Figure 5-6. In this control mode the first step is to compare the total maximum steam output of all the engines with the actual load. If the steam produced is larger than the load, then the engines are run at the fractional load factor given by the ratio of the steam load to the total steam generating capacity of the system. If the steam load is larger than the available steam, then the auto-extraction steam turbine is modulated and the auxiliary boiler is used to meet the shortfall in the steam. The next step is to compare the electrical load with the total electric output. If the load is smaller than the total output, the peak engines are shut off. If the load is still smaller, the auto-extraction turbine is modulated to lower the electrical output while keeping the extracted steam the same level. If the load is still smaller, the excess electricity is sold to the utility.

The performance of the system for the specified control mode is calculated on an hourly basis. First, the system performance is calculated for all the input load profiles for the year. This is followed by the calculation of the annual costs. In the calculation of the annual costs, the monthly costs are first computed. This is done by computing the cost for the representative working day and the non-working day of the month first, and then multiplying these costs by the appropriate number of working and non-working days of the month. These costs include the fuel costs, purchased electricity costs, and O&M costs. Also added to the monthly costs are electric demand charge and the fixed O&M costs. The annual cost is obtained by summing up the monthly costs for the year. The life-cycle operating costs are calculated using the annual cost, the present year, the year the installation is complete, and the life-cycle period of the system.

The CELCAP model has two modes for printing the results of the performance calculation. The first mode is called the brief printout mode. In this mode the following summary results are printed: important information from the input data, the monthly annual summary data on the fuel consumption and electricity generated by each engine, electricity purchased, annual fuel and O&M costs for each engine, and the component and total life cycle operating costs for each year of the life cycle period. The second printing mode is called the detailed printout mode. In this mode, the hourly peformance calculations are printed in addition to the summary results. These include the design electrical output and exported steam for each engine in the system; electrical and steam load for working and non-working days of each month; maximum hourly total output and fuel consumption of the cogeneration plant for each month of the year; and steam exported (transported out), auxiliary boiler fuel consumption, and purchased electricity numbers for working days and non-working days for each month of the year. Also in the detailed printout is the hourly plot of steam and electric demand and production of working and non-working days for each month of the year.

B. DESCRIPTION OF PRINCIPAL FUNCTIONS

The CELCAP model has several units within its structure that have specific functions. The major units in the model relate to the input data, load analysis, control mode, the engine performance calculation, and the cost escalation calculations. All these units have specific roles in the overall CELCAP model and are needed for the evaluation of a cogeneration system. The organization and description of these models are given in Sections II, III, and IV. Also given in these sections are the block diagrams of the models of these units.

The input data model is organized to read all the input data that is required for the evaluation of the candidate cogeneration system. This unit reads the data on various engine performance numbers, conditions of the steam generated, user's energy demand, and the economic parameters. Through the use of this unit, the CELCAP model can easily evaluate different cogeneration options for a given user. All that is needed for this is to change the input data on the engines. Similarly, different users can be evaluated for a given cogeneration system by merely changing the input data on the user's electrical and steam

demand. By changing the input data on economic parameters, various fuel price and escalation rate scenarios can be evaluated for the candidate system. The input data unit has a provision for specifying the control mode in which the system can operate.

The load analysis unit of the CELCAP model processes the input data. Here the actual limiting electrical and steam outputs are calculated. The total electrical and steam outputs are compared with the actual loads. Depending on the control mode chosen, the CELCAP model simulates the operation of the system accordingly and calculates the performance.

The CELCAP model has provisions for evaluating the candidate cogeneration operating in three different control modes. These three control modes are described earlier in this report. There are three procedures built in the CELCAP model so that the system can be operated according to the control mode desired. The control modes deal with how the engines in the system operate to produce electricity and steam. The three control modes are (1) engines operating at their full capacity, (2) engines operating in a mode where they follow the steam load up to their capacity, and (3) engines operating in a mode where they follow electrical load up to their capacity. The function of this provision in the CELCAP model is to make sure that the most efficient way of operating the system can be explored. For instance, a specific cogeneration system, operating at full capacity, may be more efficient than operating in the steam following control mode, whereas for some other system the situation may be reversed.

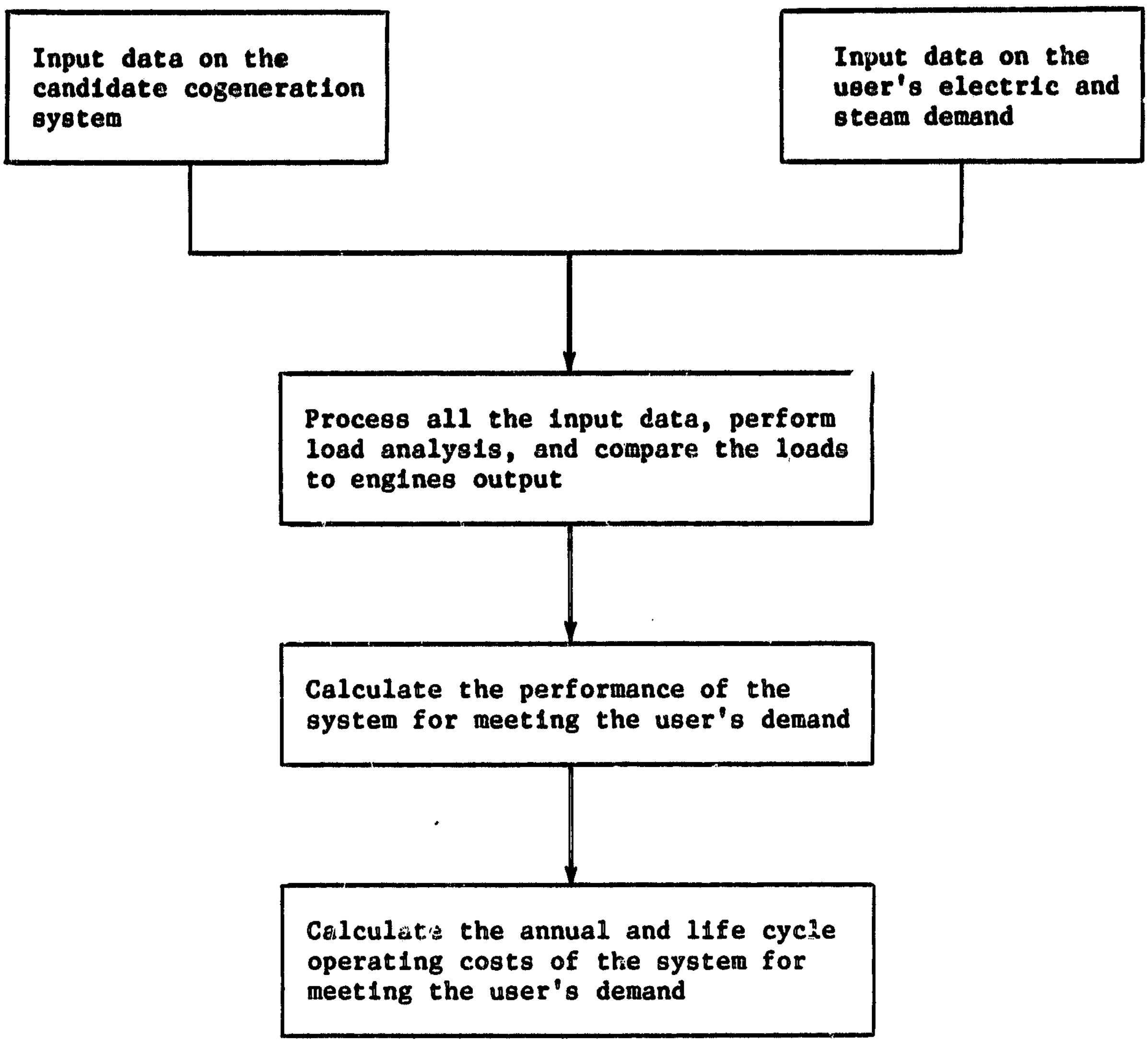

Figure 5-1. Overall Block Diagram of the CELCAP Computer Model

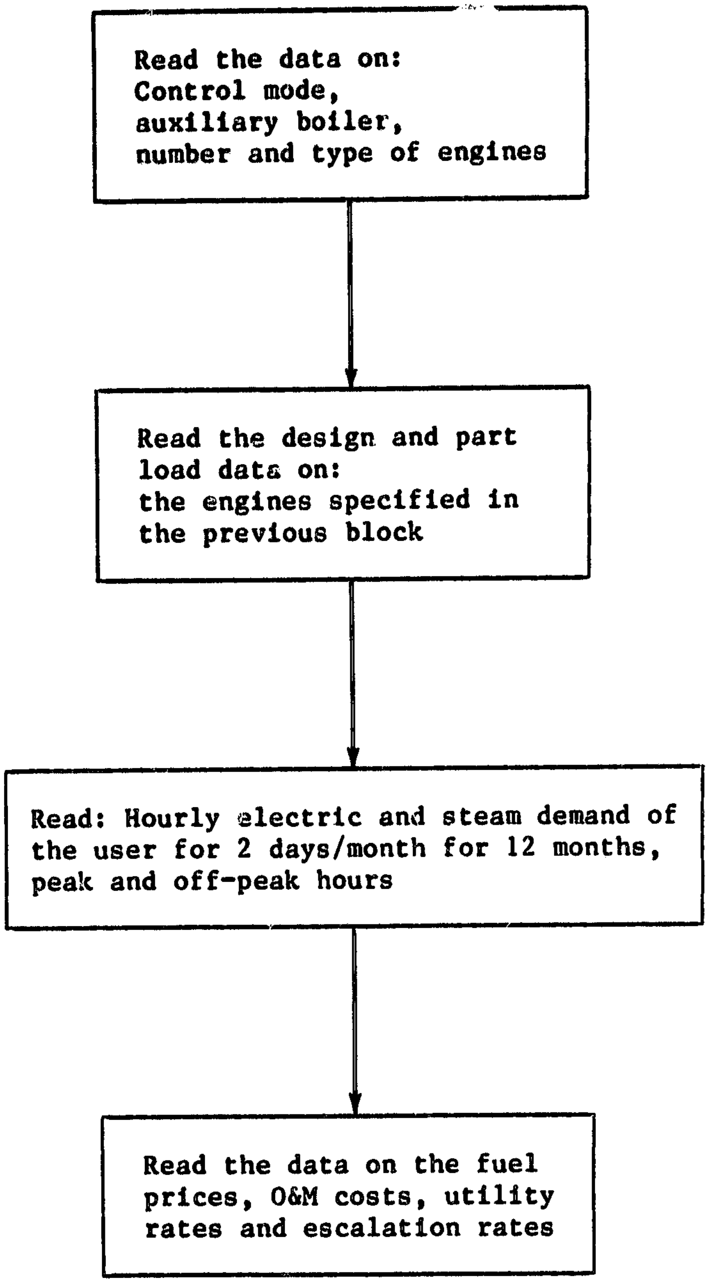

Figure 5-2. Block Diagram of the Input Data Unit in the CELCAP Model

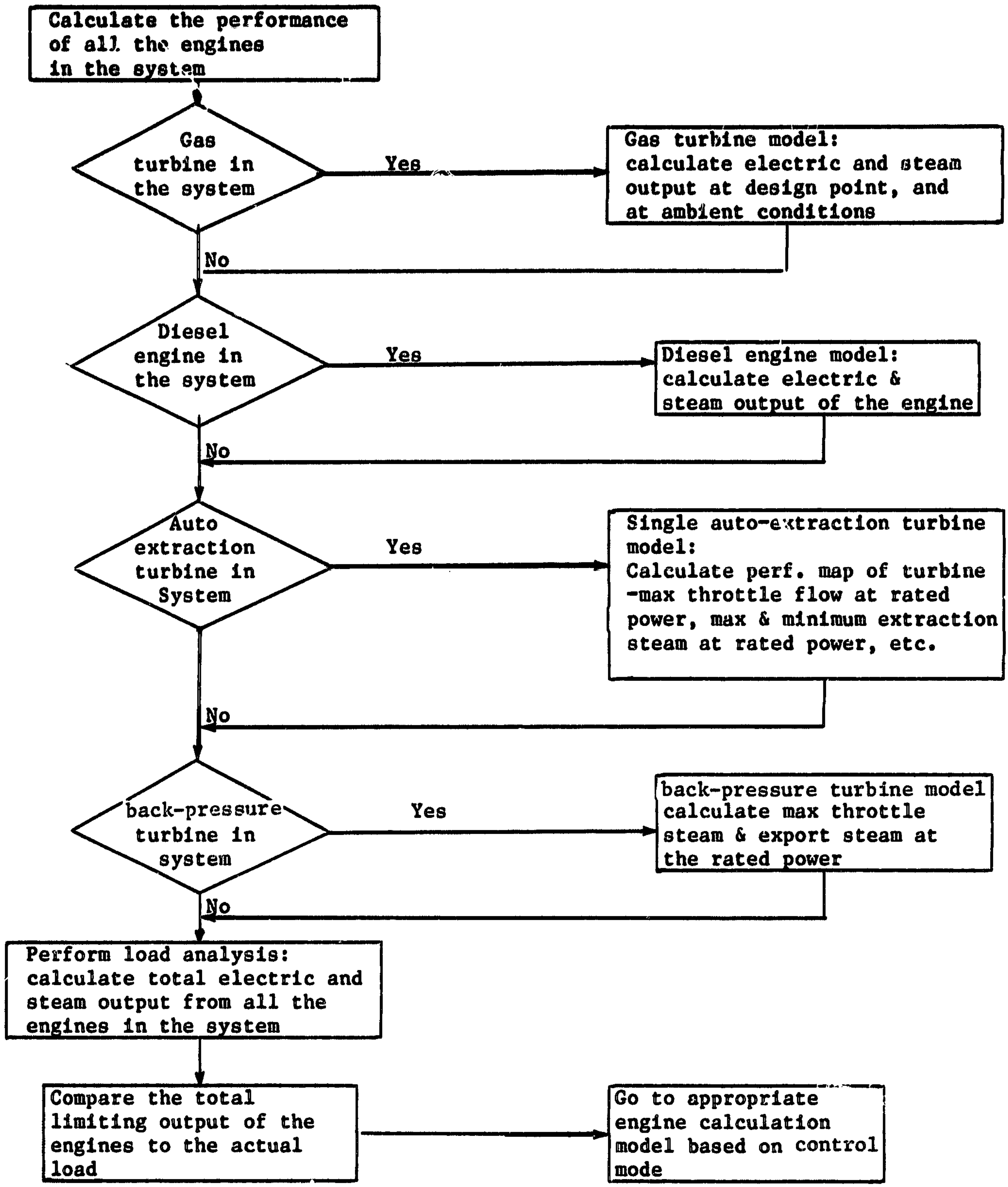

Figure 5-3. Block Diagram of the Load Analysis Unit in the CELCAP Model

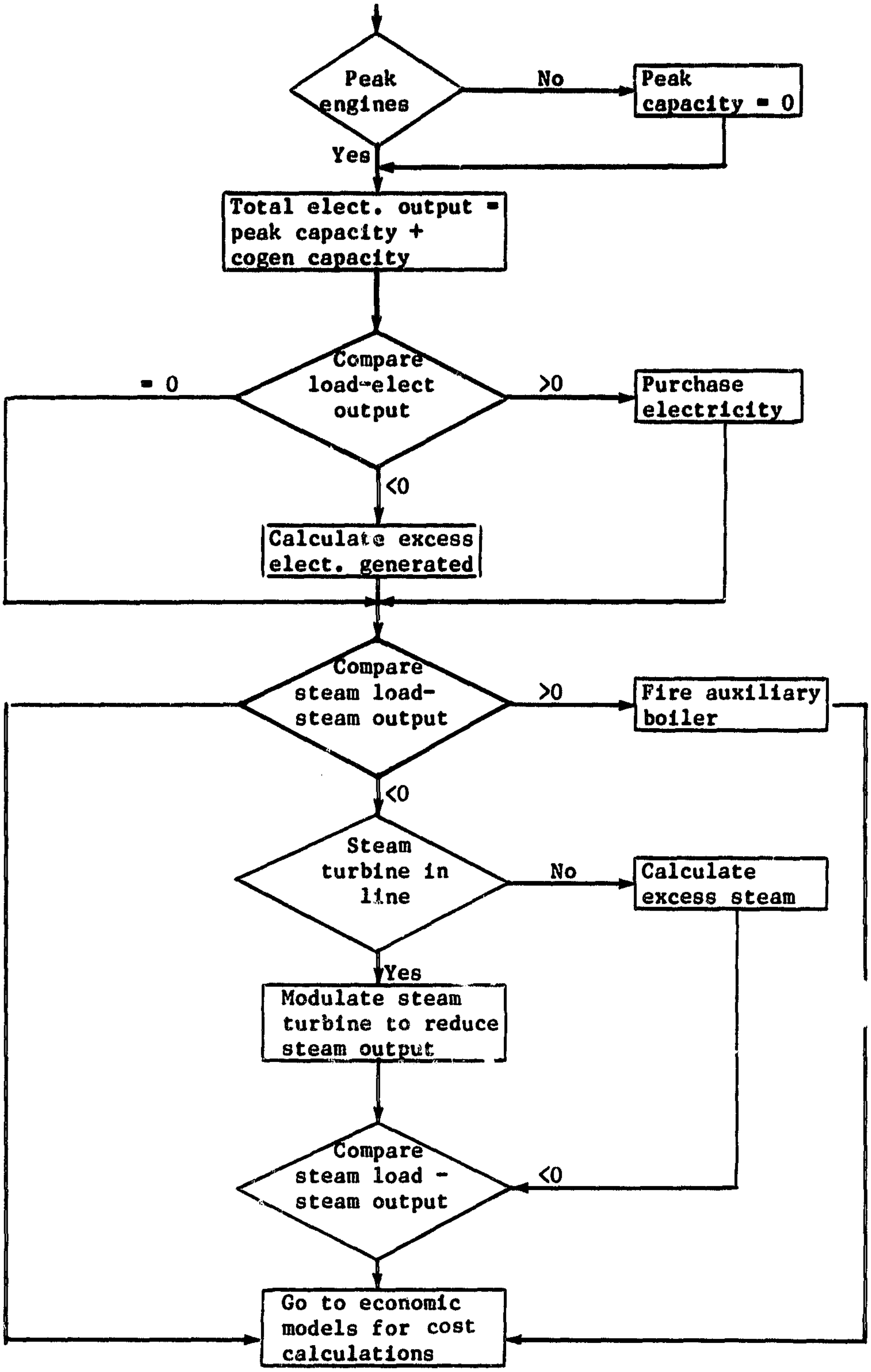

Figure 5-4. Block Diagram of Performance Calculation in the CELCAP Model for the Control Mode in Which the Engines Operate at Their Full Capacity

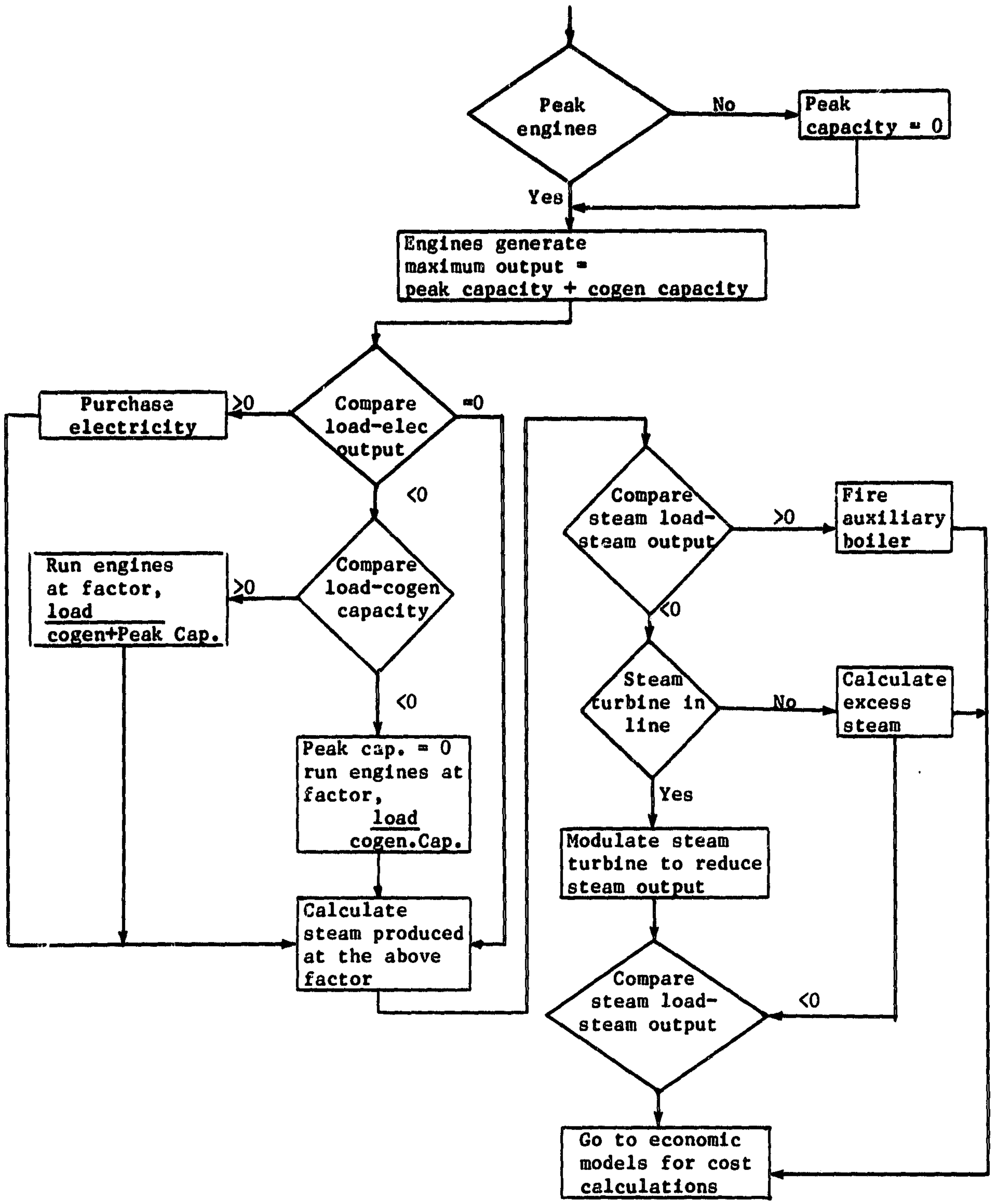

Figure 5-5. Block Diagram of Performance Calculations in the CELCAP Model for the Control Mode in Which the Engines Follow Electrical Load up to Their Capacity

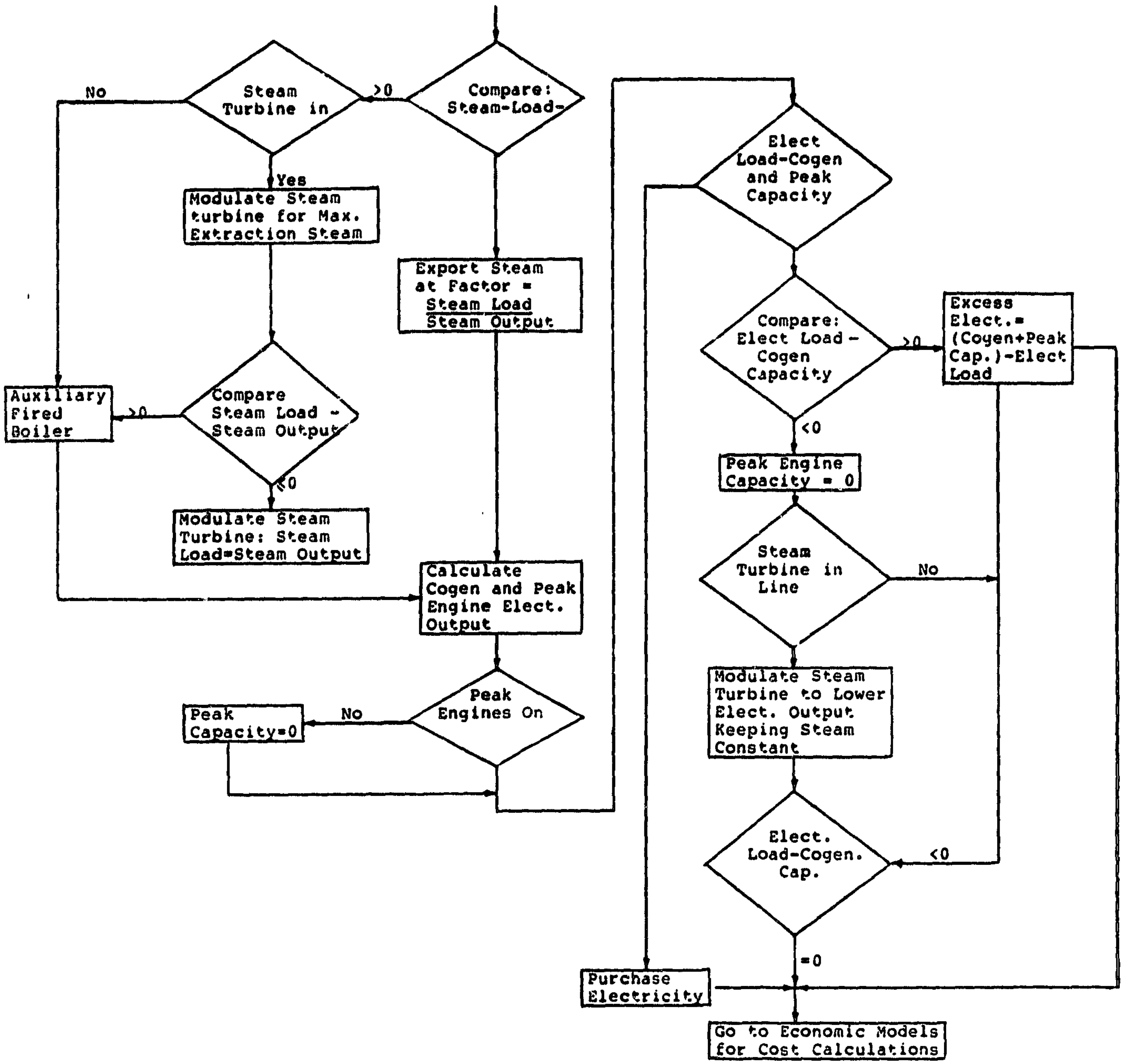

Figure 5-6. Block Diagram of the Performance Calculation in the CELCAP Model for the Control Mode in Which the Engines Follow Steam Load Up to Their Capacity

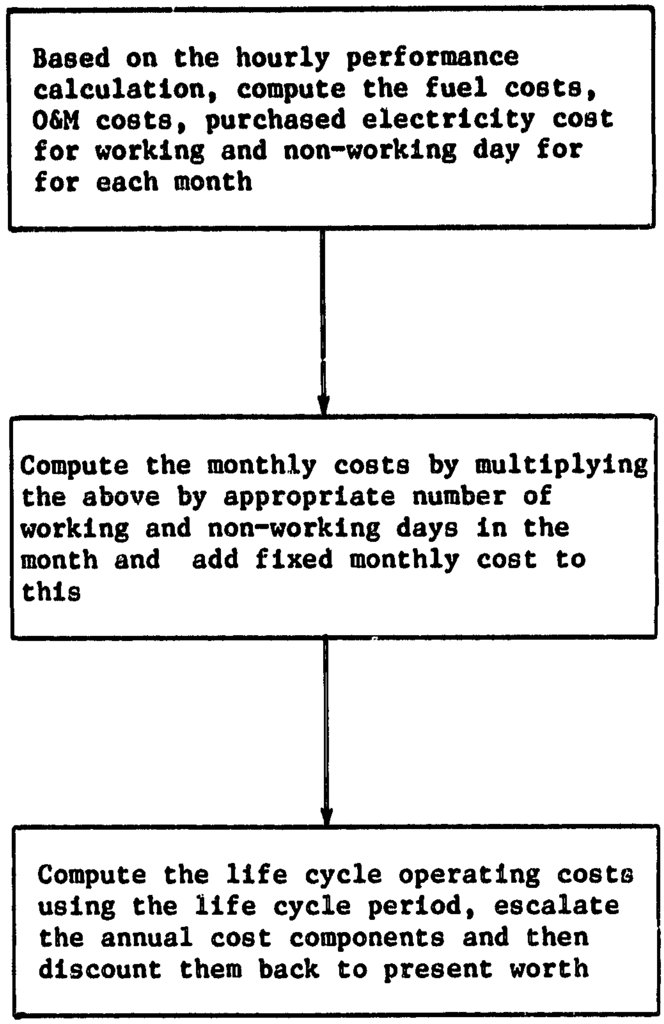

Figure 5-7. Block Diagram of the Economic Calculations Used in the CELCAP Model

SECTION VI

MODEL APPLICATIONS

There is a wide range of applications where the CELCAP model can be used for evaluating the cogeneration systems. Some of the applications where it has been successfully used are (1) selection of cogeneration system capacity for a Naval center, (2) modification of the on-site cogeneration system at a Naval base, (3) optimizing the operation of a cogeneration system at a Marine Corps base, (4) evaluating the effects of fuel switching on the operating cost of a cogeneration system, and (5) effect of changing the steam distribution system on the operating cost of the cogeneration plant.

In one of the earliest studies using the CELCAP model, Cooper examined the cogeneration options for the Naval hospital at Beaufort, South Carolina. He evaluated cogeneration options consisting of one and two gas turbines and one and two diesel engines. In another study, Lee and Cooper (Reference 5) used the CELCAP model to evaluate the cogeneration system at a Naval base for an air-conditioning switch-over from absorption type to electric type during the summer months. Three detailed examples of the CELCAP model applications are described in this section. These are (1) cogeneration system selection, (2) cogeneration system modification, and (3) cogeneration system optimization.

A. COGENERATION SYSTEM SELECTION

In this category of applications, the CELCAP model is used for selecting the optimum type and capacity for the cogeneration system for a user's facility. This may be an existing facility without a cogeneration system or one to be constructed. Lee (see Reference 7) used the CELCAP model to evaluate the cogeneration potential at the Naval Construction Battalion Center, Port Hueneme, California. The cogeneration options he examined were single and multiple combinations of gas turbines with waste heat boilers with a total capacity of 1500 to 5600 kW and single and multiple combinations of diesel engines with waste heat boilers with a total capacity of 2000 kW to 5000 kW. Lee evaluated all these options for the existing electric and steam loads at the facility and recommended the ones that had the shortest payback periods.

Table 6-1 is a list of all the cogeneration options Lee evaluated in his study. Typical schematics of the cogeneration system are shown in Figure 6-1 for the gas turbine arrangement and Figure 6-2 for the diesel engine arrangement. The data on the engine performance that were required to be input to the CELCAP model are shown in Tables 6-2 and 6-3 for gas turbine and diesel engines, respectively. The hourly profiles of electric and steam demands of the facility for typical winter and summer months are shown in Figures 6-3 and 6-4. The data on fuel and purchased electricity prices, operation and maintenance costs of various engines, and the various escalation rates are shown in Table 6-4. All these data are required by the CELCAP model to calculate the annual and life-cycle operating costs of the various cogeneration

options. Using the capital cost data on the engines shown in Table 6-5, Lee computed the payback periods and SIR numbers for all the cogeneration options. The economic results of this evaluation are shown in Tables 6-6 and 6-7 for several cogeneration options involving gas turbines and diesel engines. Lee recommended the best cogeneration option based on the results shown in Table 6-8.

B. COGENERATION SYSTEM MODIFICATION STUDIES

In this type of application, the CELCAP model is used for evaluating the modifications on an existing cogeneration system. The modifications can be major ones such as replacing all the existing engines with new engines of different type and capacity or minor ones such as switching the fuel used in boilers from natural gas to fuel oil. In a study on the cogeneration options for a Naval base, Birur (see Reference 11) used the CELCAP model to evaluate several modifications on the existing cogeneration system at the base. Based on this evaluation, Birur made recommendations on the modifications that had the lowest operating costs.

The existing cogeneration system at the base consists of three auto-extraction steam turbines and four boilers. A schematic of the existing steam is shown in Figure 6-5. The steam Turbines T3, T4, and T5 generate 3.5 MW, 5 MW, and 4 MW, respectively. All the steam turbines exhaust to condensers at 1 1/2 in. Hga. The steam is extracted from Turbines T3 and T5 at 200 psig, whereas it is extracted at 5 psig from Turbine T4. The 200-psig extracted steam is exported from the power plant to the various buildings on the base through the base steam distribution system.

Several cogeneration arrangements were evaluated for the base using the CELCAP model. A list of these arrangements is shown in Table 6-9. Some of these arrangements are modifications on the existing system while others are new systems consisting of completely new engines. All these options were evaluated for two steam distribution pressures - 200 psig and 125 psig. The data on the engines in the existing system are shown in Table 6-10. The data for the new engines are shown in Tables 6-11 and 6-12. The fuel and purchased electricity prices, O&M costs, and their escalation rates are shown in Table 6-13.

These data were used along with the hourly electric and steam demand profiles of the base to evaluate the performance of the various cogeneration arrangements. In Table 6-14 the results of this evaluation are shown for a few cogeneration options. Using the capital cost of the new equipment needed for the options, the payback period and the SIR ratio are calculated. This result is shown in Table 6-15 for some of the cogeneration options.

C. COGENERATION SYSTEM OPTIMIZATION

In this type of application, the CELCAP model is used for performing optimization studies on the operations of an existing cogeneration system. A cogeneration system with several engines has a wide flexibility in operation.

Apart from the choice of running or not running an engine at any given hour, the number of engines to be run can also be varied. The ratio of electric-to-steam loads and purchased electricity prices also dictates, to some extent, the efficient way of running the on-site engines. Because of these multiple system options, the ideal choice is not clear. The CELCAP model, through its ability to simulate the operation of the cogeneration system, can provide optimization studies leading to optimum plant operation. An example of such a study is given below.

The CELCAP model was used to perform an optimization study on a cogeneration system at a Marine base in South Carolina. The existing system at the base consisted of three auto-extraction steam turbine/generators of 1000 kW each, three boilers supplying steam for the turbines, and one auxiliary boiler. A schematic of the system is shown in Figure 6-6. The cogeneration system supplies the electricity and steam to meet the demand of the base. While all of the steam supplied is generated on-site, some of the electricity supplied by the cogeneration system is purchased from the local utility company. The electric demand of the base is mainly for lighting, small machinery, and air-conditioning needs. The steam demand of the base is for the space and water heating, cooking, and absorption air-conditioning needs. The two major objectives of the optimization study were to examine the impact on the operating cost of the system due to changing (1) the purchase/generation ratio for electricity, and (2) the ratio of absorption air conditioning to electric air conditioning.

Four engine arrangement options were considered for the evaluation. The first, second, and third arrangements consisted of three, two, and one engine(s), respectively. The fourth arrangement did not have any engine at all. In all these options, only one turbine was used in the extraction mode and run all hours year round. Depending on the arrangement, the second and third turbines were run as peaking units in the daytime during summer months. Each of these cogeneration options was evaluated for several cases of peak-hour operations. The electric and steam load data of the base for five ratios of absorption-to-electric air conditioning were constructed and the cogeneration options were evaluated for all the air-conditioning ratios.

In Figures 6-7 and 6-8, electric and steam demand profiles for a typical working day in March are shown. Typical results from the evaluation are shown in Table 6-16 for the case of the two-turbine arrangement of the cogeneration system. The results from the evaluation were carefully analyzed to select the most economical cogeneration system for the base. The results also show the best ratio of absorption to electric air conditioning for the selected cogeneration option.

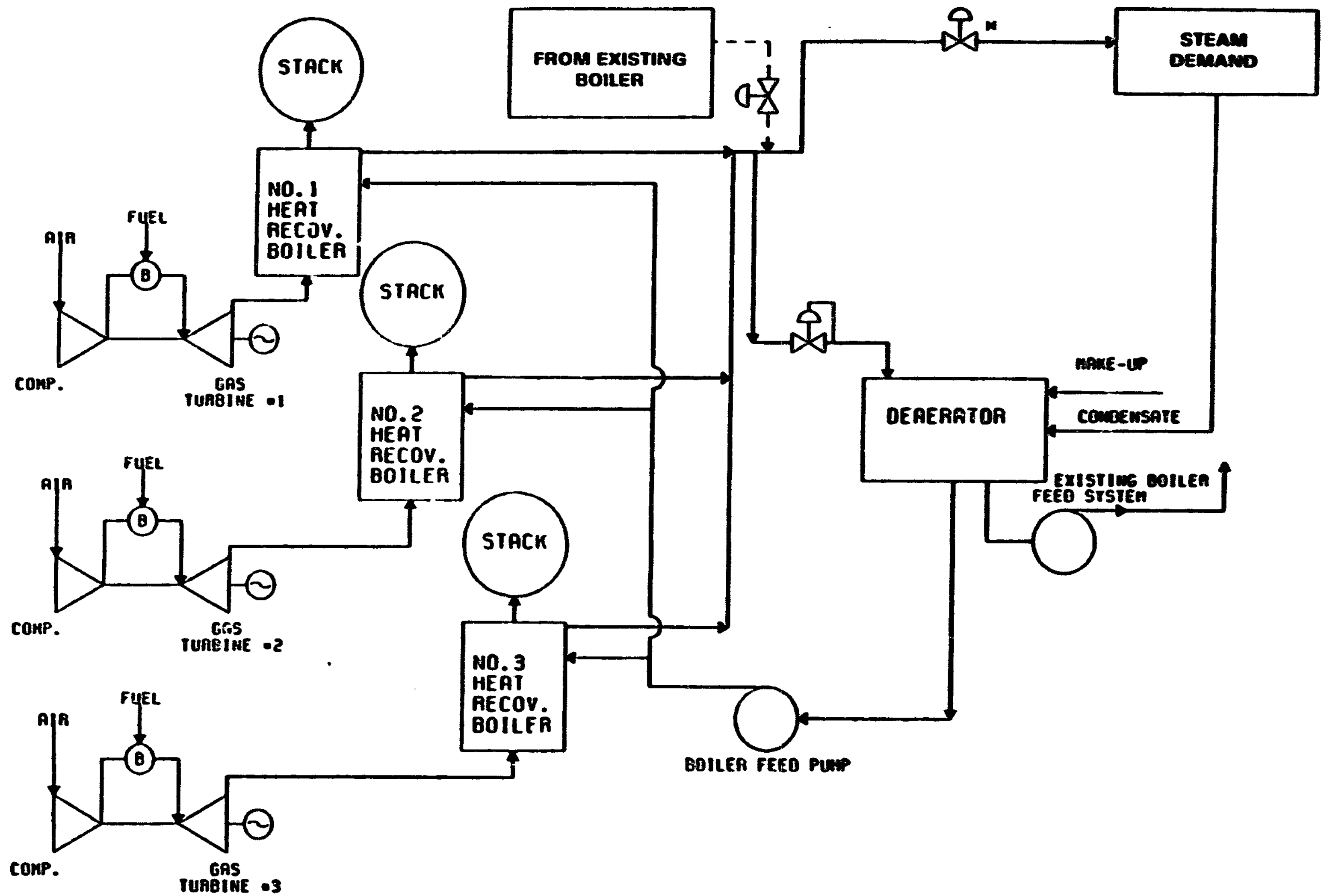

Figure 6-1. Gas Turbine Generators with Unfired Heat Recovery Boilers

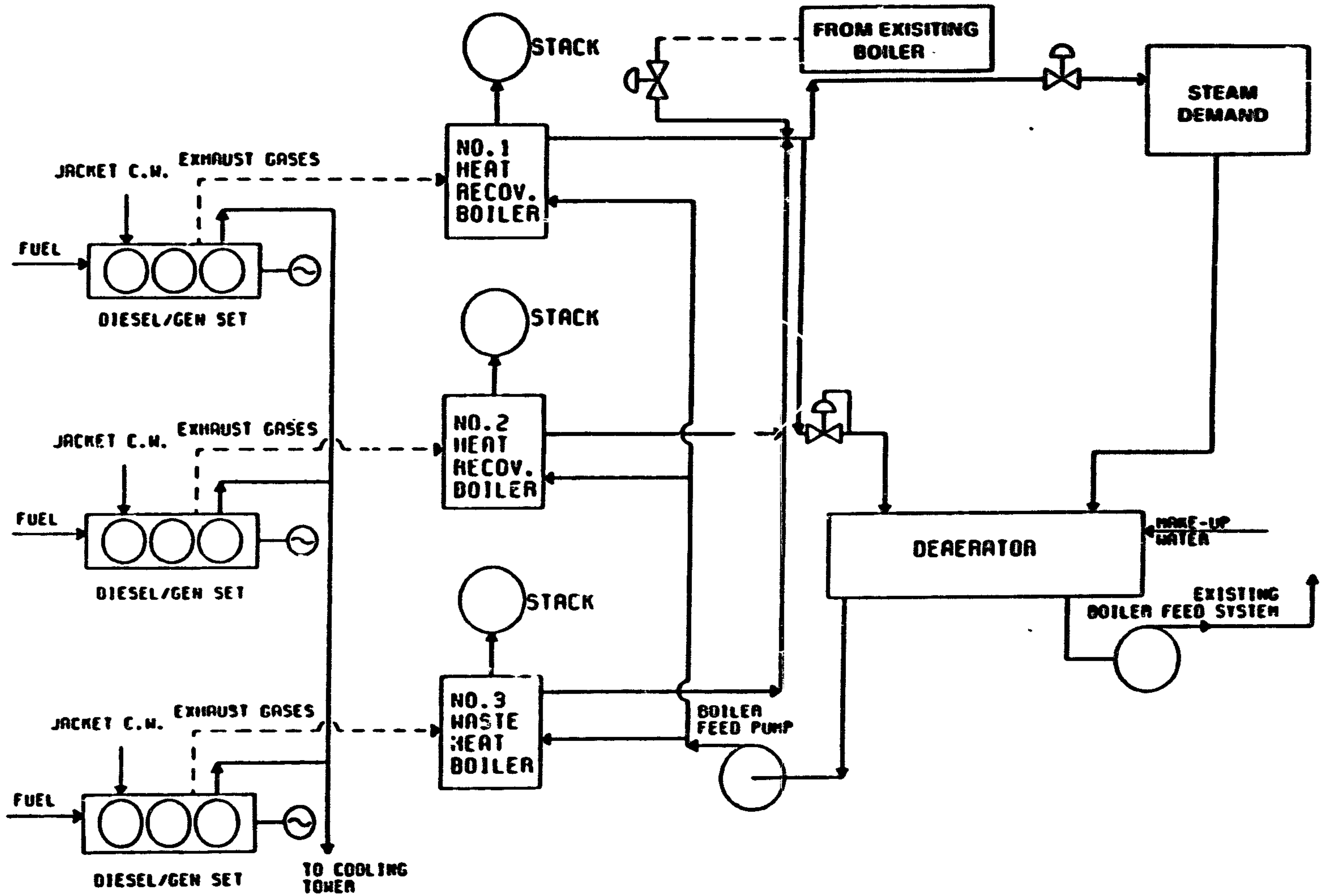

Figure 6-2. Diesel Generator Units with Heat Recovery Boilers

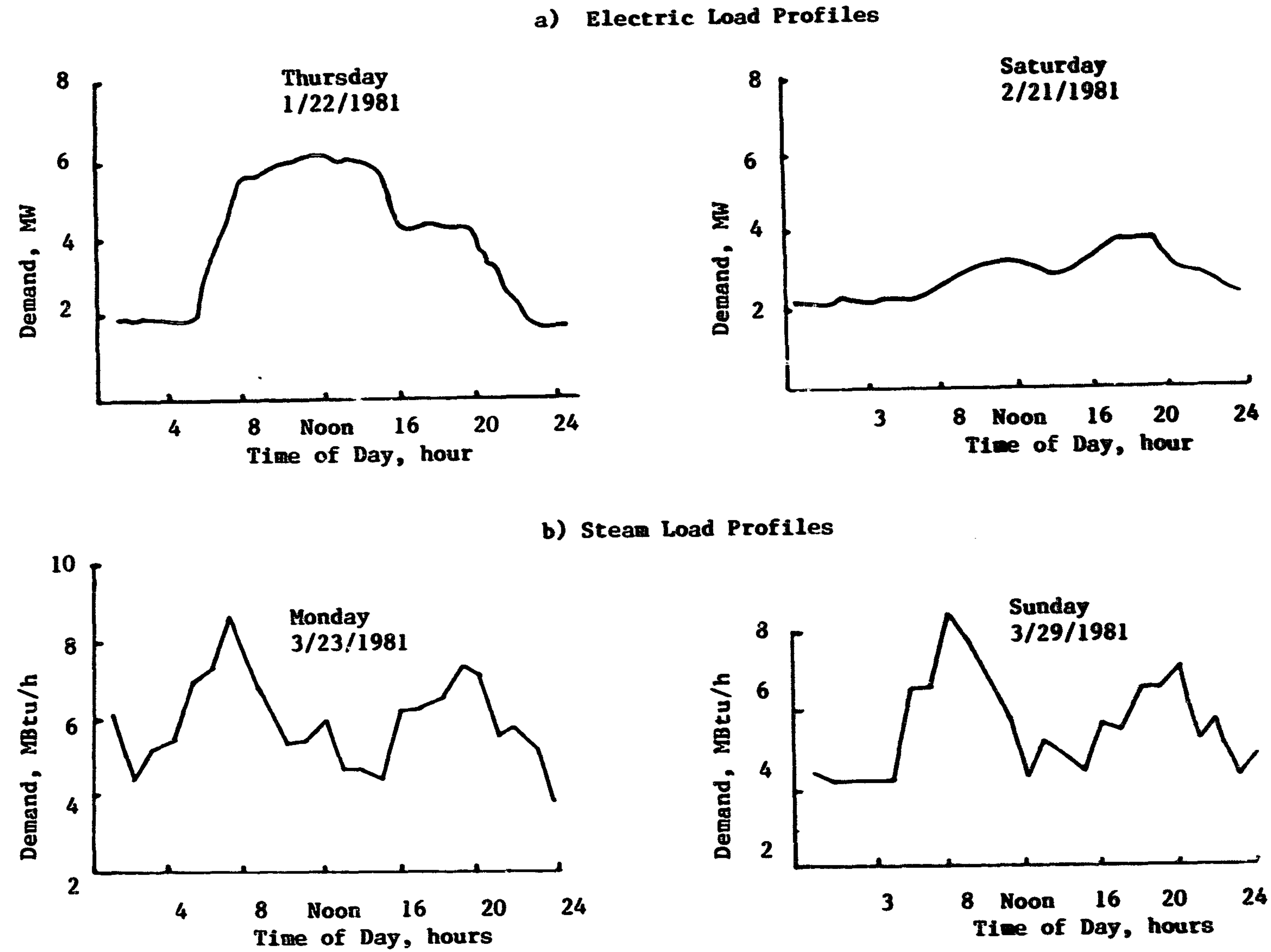

Figure 6-3. Typical Electric and Steam Load Profiles of Working and Non-Working Days for a Winter Month

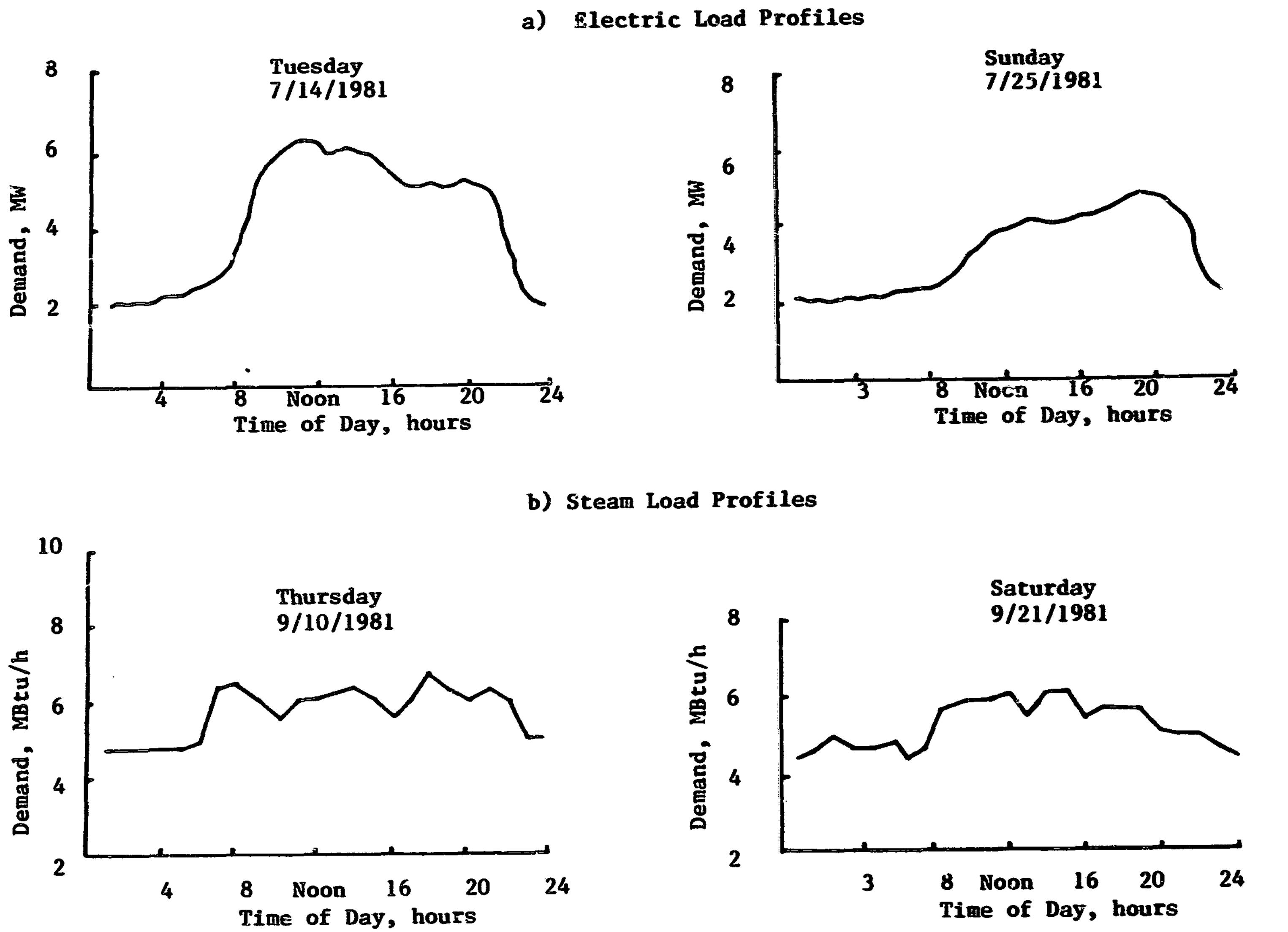

Figure 6-4. Typical Electric and Steam Load Profiles of Working and Non-Working Days for a Summer Month

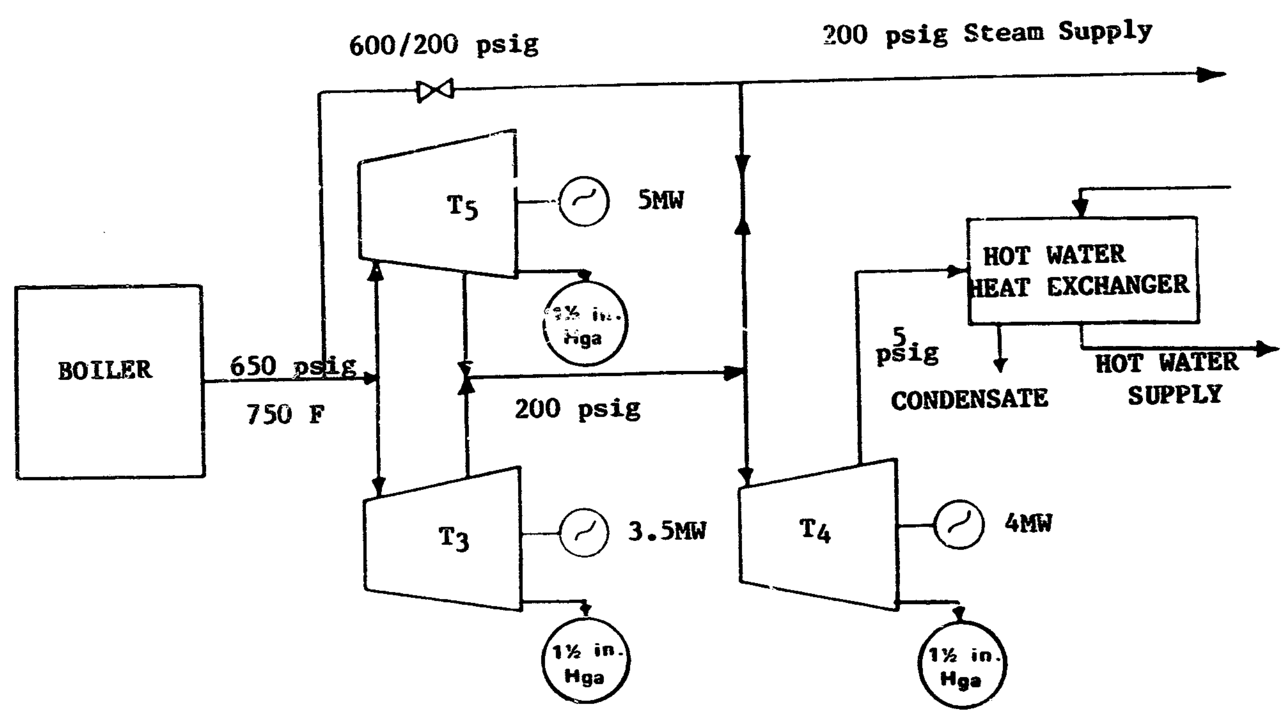

Figure 6-5. Schematic of Existing Cogeneration System at the Navai Base

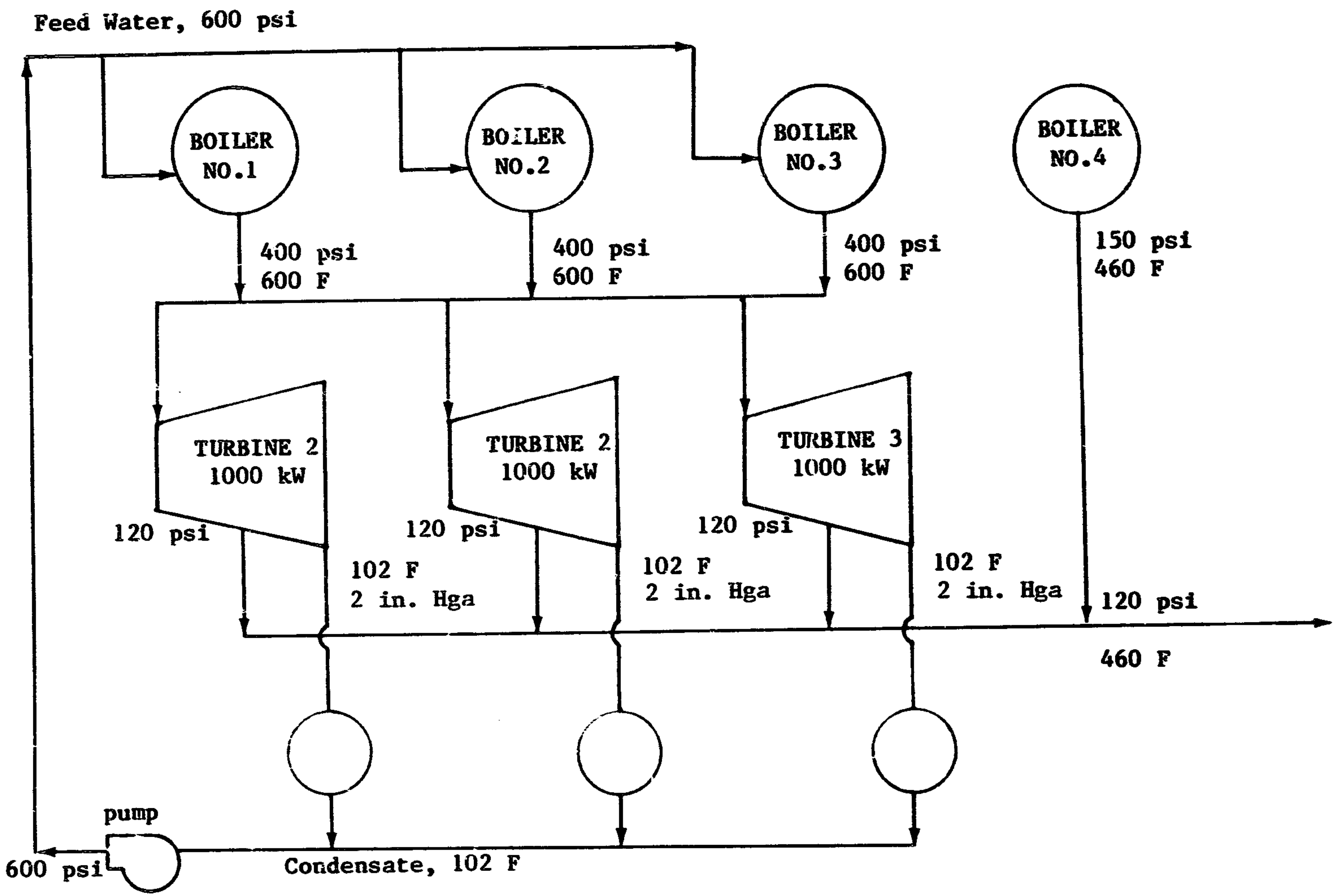

Figure 6-6. Schematic of the Central Power Plant at MCRD, Parris Island

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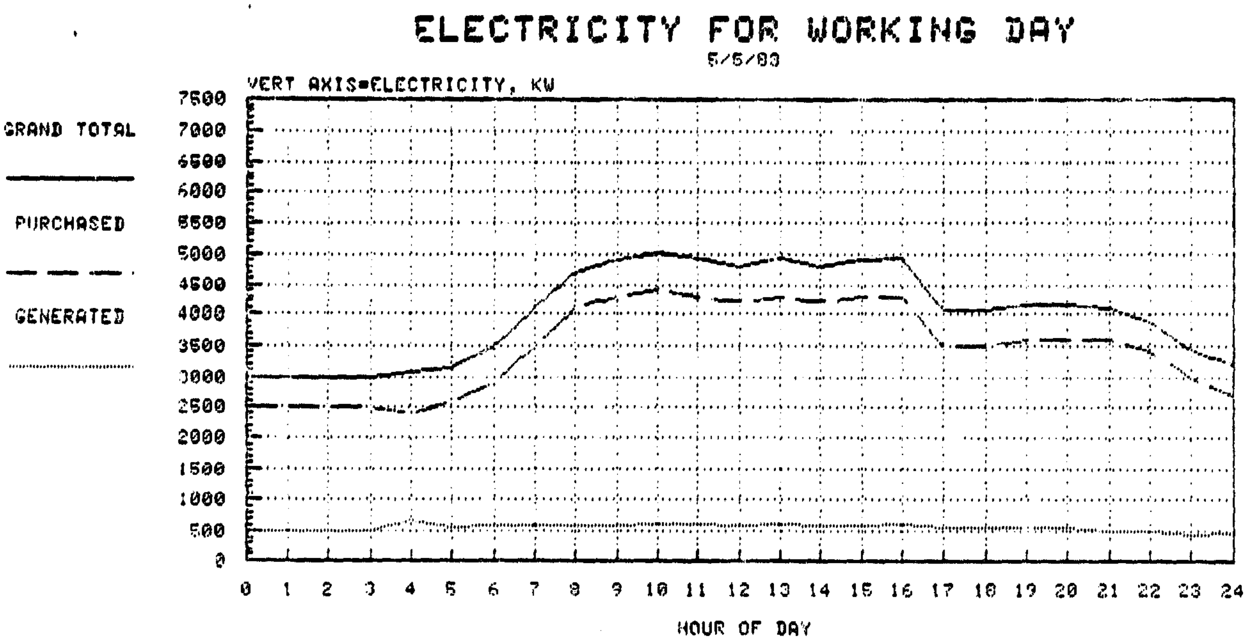

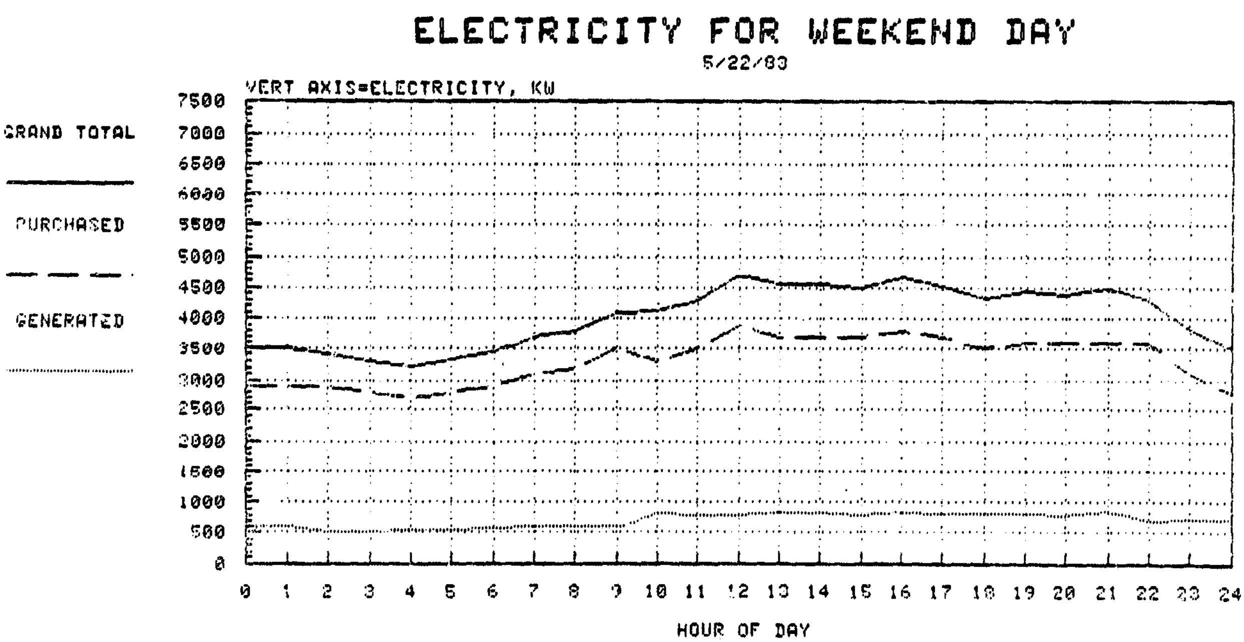

Figure 6-7. Typical Electric Demand Profile of MCRD for the Month of May

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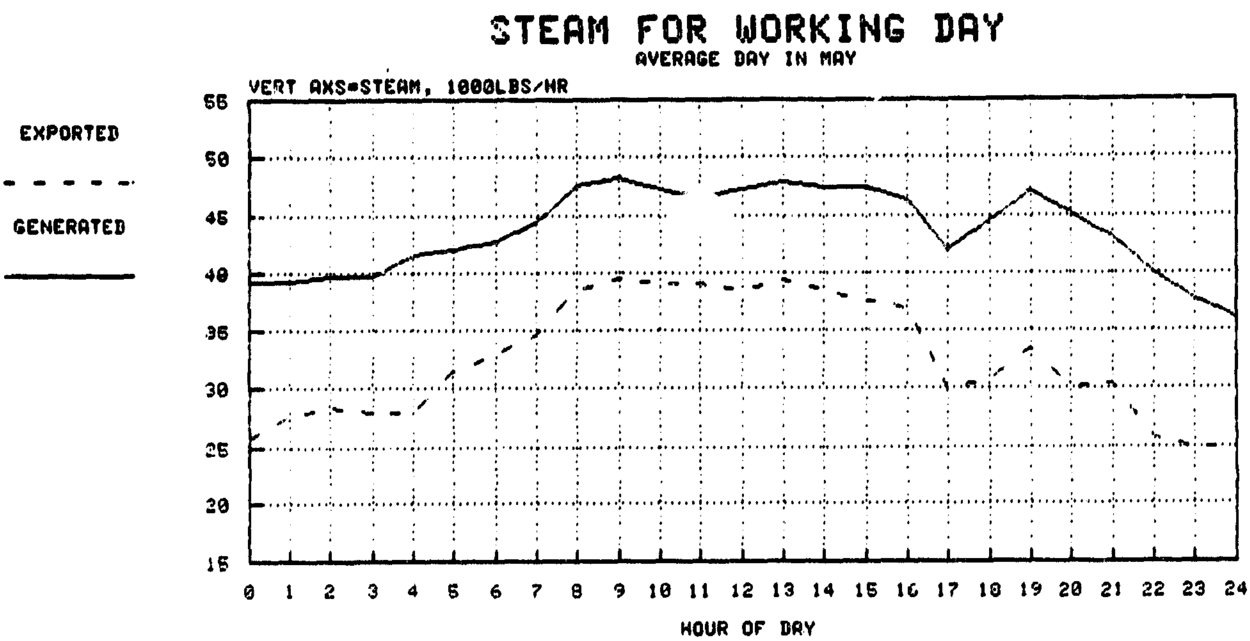

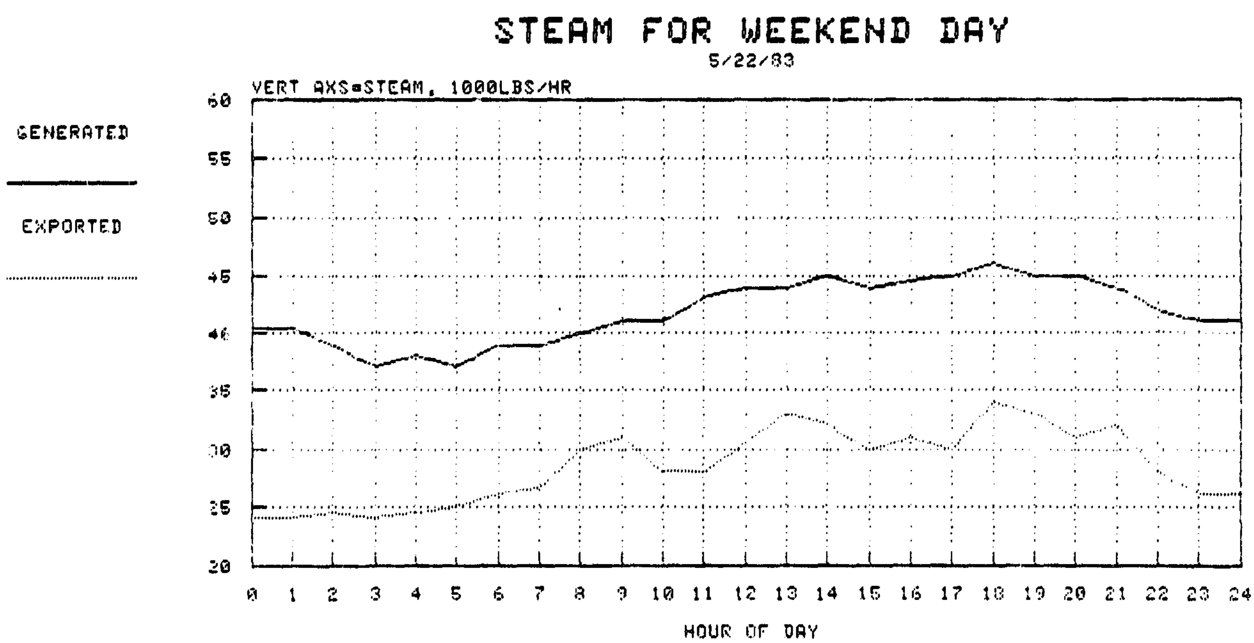

Figure 6-8. Typical Steam Demand Profile of MCRD for the Month of May

Table 6-1. Cogeneration Options Evaluated

Concept 1	**Gas Turbine-Generators With Waste Heat Recovery Boilers**
Option 1G:	Three 500-kW gas turbine generator sets
Option 2G:	Two 800-kW gas turbine generator sets
Option 3G:	Three 800-kW gas turbine generator sets
Option 4G:	One 2800-kW gas turbine generator set
Option 5G:	Two 2800-kW gas turbine generator sets
Concept 2	**Diesel Generator Sets with Waste Heat Recovery Boilers**
Option 1D:	Two 1000-kW diesel generators
Option 2D:	Three 1000-kW diesel generators
Option 3D:	One 2500-kW diesel generators
Option 4D:	Two 2500-kW diesel generators
Option 5D:	One 4000-kW diesel generators

Table 6-2. Design Point Data for Gas Turbines and Diesel Engines

(a) Gas Turbines

Gas Turbine	Rated Capacity, kW	Standard Pressure, psia	Standard Temperature, °F	Fuel Flow, Btu/h	Air Flow, lbm/h
Garrett 1E831-800	515	14.7	59	8,250,000	28,050
Solar Saturn GSC-4000	800	14.7	59	13,000,000	49,1590
Solar Centaur	2,780	14.7	59	39,052,631	138,036

(b) Diesel Engines

Diesel Engine	Rated Capacity, kW	Fuel Consumption, Btu/h		Exhaust Gas Temperature, °F		Total Exhaust Flow, lbm/h
		Full Load	Partial Load	Full Load	Partial Load	
Cummins KTA-3067-CC	1000	8,400,000	N/A	925	N/A	14,062
Cooper-Bessemer KSV 450, 16 cyl	2480	23,560,000	17,632,000 (3/4 load)	810	763	42,840
Cooper-Bessemer	4130	38,608,000	29,184,000 (3/4 load)	900	850	62,032

Table 6-3. Energy Price, O&M Costs, and Escalation Rates

Electrical Power:

(based on current schedule No. TOU-8 of Southern California Edison Company)

Customer Charge, $/month	560.00
Fuel Adjustment Charge, $/kWh	0.0540

	Demand Charge, $/kW	Energy Charge, $/kWh
On-Peak	4.05	0.01256
Mid-Peak	0.65	0.00919
Off-Peak	0.00	0.00583

Daily time periods based on Pacific Standard Time are defined as follows:

On-peak:	12:00 noon to 6:00 p.m., summer weekdays except holidays
Mid-peak:	8:00 a.m. to 12:00 noon and 6:00 p.m. to 10:00 p.m. summer weekdays except holidays
Off-peak:	All other hours
Summer months:	May to October

Boiler Fuel:

Natural Gas	$3.96/MBtu

O&M Costs:

Natural Gas Fired Boilers:	$1.35 per thousand pounds of steam generated
Waste Heat Recovery Boilers:	$1.10 per thousand pounds of steam generated
Gas Turbine/Generators:	$4.00 per MWH power generated
Diesel Engine Generators:	$13.00 per MWH power generated

The boiler efficiencies for the existing and new boilers were estimated to be 68% and 58%, respectively.

Table 6-3. Energy Price, O&M Costs, and Escalation Rates (Cont'd)

	Escalation Rates during		Present Worth Factor
	1982-1985	1990-2010	1985-2010
Natural Gas	14.0	8.0	20.050
Electricity	13.0	7.0	18.049
O&M	5.6	0.0	9.524
Discount Rate	10%		

DOE RATES, %
(DOE Region 9: Industrial Sector)

	Escalation Rates during			Present Worth Factor
	1982-1985	1990-2010	1985-2010	1985-2010
Natural Gas	8.87	-0.77	0.98	15.93
Electricity	5.29	-0.53	-0.91	13.40
O&M[a]	0	0	0	
Discount Rate		7%		

Modified DOE Rates, %, for CELCAP Input

	Escalation Rates during		Present Worth Factor
	1982-1985	1985-2010	1985-2010
Natural Gas	8.87	-0.10	11.952
Electricity	5.29	-1.15	10.886
O&M[a]	0.0	0.0	12.061
Discount Rate	7%		

[a]Estimated

Table 6-4. Estimated Capital Costs for Various Cogeneration Systems

	Gas Turbines			Diesel Engines		
	Garrett IE831-800	Solar Saturn GSC-1200	Solar Centaur GSC-4000	Cummins KTA-2067-GC	Cooper-Bessemer KXV-450	Cooper-Bessemer KSV-450
Rated Capacity, kW	515	800	2,780	1,000	2,480	4,130
Generator Set/Boiler, $	449,000	700,000	1,320,000	499,000	1,272,000	2,135,000
Installation, $	135,000	210,000	396,000	150,000	381,000	640,000
Miscellaneous,[a] $	199,000	309,000	583,000	221,000	562,000	945,000
Total cost, $	783,000	1,220,000	2,300,000	870,000	2,215,000	3,720,000

[a]Miscellaneous includes: 9% engineering, 15% management and profit, and 10% contingency.

Table 6-5. Economic Results for Gas Turbine Cogeneration Concepts Using NAVFAC Escalation Rates, Dollars x 10^3

Case No.	1R	1G	2G	3G	4G	5G
Cogeneration Option	None New Boilers	3 x 500 kW	2 x 800 kW	3 x 800 kW	1 x 2800 kW	2 x 2800 kW
Operation Mode[a]	-	1	1	1	1	1
Purchased Utility Peak On/Mid/Off	5800/7140/5600	4340/5660/4100	4280/5600/4010	3500/4800/3220	3200/4460/2860	700/1800/360
Revenue from Sale of Excess Electricity	0	0	0	0	0	0
Annual Energy Consumption MBtu, x 10^3	563	502	508	529	491	564
Total First Annual Operation Cost	3,415	2,955	2,971	3,006	2,786	2,892
Total First Annual Savings	-	460	444	49	629	523
Total Capital Costs	1,000	2,350	2,440	3,660	2,300	4,600
Total 25-yr Life-Cycle Costs	62,658	54,606	54,985	55,869	51,812	54,530
Discounted Savings to Investment Ratio (SIR)	-	5.96	5.33	2.55	8.34	2.26
Energy Savings to Cost Ratio	-	45.2	38.2	12.8	55.4	-0.3
Simple Payback Period, yr	-	2.9	3.2	6.5	2.1	6.9

[a]Engine runs at the rated capacity.

Table 6-6. Economic Results for Diesel Turbine Cogeneration Concepts Using NAVFAC Escalation Rates, Dollars x 10^3

Case No.	1R	1D	2D	3D	4D	5D
Cogeneration Option	None New Boilers	2 x 1000 kW	3 x 1000 kW	1 x 2500 kW	2 x 2500 kW	2 x 4000 kW
Operation Mode[a]	-	2	2	2	2	2
Purchased Utility Peak On/Mid/Off	5800/7140/5600	3800/5140/3600	2800/4140/2600	3310/4600/3110	900/2170/630	1600/340/1460
Revenue from Sale of Excess Electricity	0	0	115.65	27.53	674.32	400.53
Annual Energy Consumption MBtu, x 10^3	563	492	461	499	523	491
Total First Annual Operation Cost	3,415	2,970	2,682	2,953	2,353	2,493
Total First Annual Savings	-	436	733	462	1,062	922
Total Capital Costs	1,000	1,740	2,610	2,215	4,430	3,720
Total 25-yr Life-Cycle Costs	62,658	54,109	48,252	53,666	41,854	44,586
Discounted Savings to Investment Ratio(ROI)	-	11.6	8.90	7.40	6.07	6.64
Energy Savings to Cost Ratio	-	95.9	63.4	52.7	11.7	26.5
Simple Payback Period, yr	-	1.7	2.2	2.6	3.2	3.0

[a]Engine runs at the rated capacity.

Table 6-7. Comparison of the Economics of the Cogeneration Concepts

Option, Case No.	Total Annual Dollar Savings	SIR	E/C	Simple Payback Period, yr
<u>Gas Turbine Concepts Using NAVFAC Rates</u>				
2800 kW (4G)	629,000	8.34	55.4	2.1
3 x 500 kW (1G)	460,000	5.96	45.2	2.9
2 x 800 kW (2G)	444,000	5.33	38.2	3.2
3 x 800 kW (3G)	409,000	2.55	12.8	6.5
2 x 2800 kW (5G)	523,000	2.26	-0.3	6.9
<u>Diesel Engine Concepts Using NAVFAC Rates</u>				
2 x 1000 kW (1D)	436,000	11.6	95.9	1.7
3 x 1000 kW (2D)	733,000	8.90	63.4	2.2
2500 kW (3D)	462,000	7.40	52.7	2.6
4000 kW (5D)	922,000	6.64	26.5	3.0
2 x 2500 kW (4D)	1,062,000	6.07	11.7	3.2

Note: For any option to be qualified for a valid military construction project, the following criteria need to be met:

(1) SIR greater than one when calculated per NAVFAC P-442.

(2) Minimum required DOD energy SIR ratio (E/C) of at least 16 for FY85.

Table 6-8. List of Cogeneration Arrangements Evaluated

Options with the existing steam distribution system:

1. The existing cogeneration system with steam Turbines T_4 and T_5 (T_5 operating as a base unit at 2000 kW).

2. A system with Turbine T_3 only.

3. A system with a back-pressure turbine (200 psig exhaust pressure) replacing Turbine T_4, and with Turbines T_3 and T_5 operating.

4. A system with a back-pressure turbine only (exhaust pressure 200 psig).

Options with the steam distribution system converted to 125 psig:

1. A system with Turbine T_5 (at output limited to 3500 kW) and a 7000-kW back-pressure steam turbine.

2. A system with steam Turbine T_3 (extraction pressure of 125 psig and exhaust pressure of 20 in. Hga).

3. Options A(3) and A(4) with an exhaust pressure of 125 psig.

4. Options A(5) with gas turbine exhaust boiler pressure of 125 psig.

Table 6-9. Characteristics of the Steam Turbines and Boilers in the Existing System

Item	Description
Turbine 3, Manufactured 1944	3500 kW, 600 psig, 725°F inlet; 1-1/2 inch Hga exhaust; 210 psig and 4 psig extraction
Turbine 4, Manufactured 1940	4000 kW, 180 psig, 480°F inlet; 1-1/2 inch Hga exhaust and 5 psig extraction
Turbine 5, Manufactured 1940	5000 kW, 600 psig, 750°F inlet; 1-1/2 inch Hga exhaust and 200 psig extraction
Boilers 1, 2, and 3	76,000 lb/h capacity at 625 psig, 750°F
Boiler 5	80,000 lb/h capacity at 625 psig, 750°F

C-2

Table 6-10. Characteristics of Back-Pressure Steam Turbines

Rated Capacity, kW	Throttle Steam condition, psig/°F	Exhaust Steam condition, psig/°F	Steam Rate, lb/kWh Full load	Steam Rate, lb/kWh Half load
15,000	600/750	200/510	37.0	44.0
10,000	600/750	200/510	38.0	45.0
7,500	600/750	200/510	39.0	46.0
5,000	600/750	200/510	40.0	47.0
3,000	600/750	200/510	42.0	48.0
15,000	600/750	125/410	27.0	31.0
10,000	600/750	125/410	27.5	31.5
7,500	600/750	125/410	28.0	32.0
5,000	600/750	125/410	29.0	33.0
3,000	600/750	125/410	30.0	34.0
750	600/750	125/410	34.0	39.0

Table 6-11. Characteristics of Gas Turbines

Gas Turbine	Rated Capacity, kW	Standard Pressure, psia	Standard Temperature, °F	Fuel Flow, 10^3 lb/h	Air Flow, 10^3 lb/h
General Electric G5261	18,900	14.7	59	252,000	767
General Electric G3142	10,150	14.7	59	138,550	408
Solar Mars	7,400	14.7	59	81,900	288
Allison 501-KB	3,338	14.7	59	42,759	132
Solar Centaur	2,780	14.7	59	39,053	138

Table 6-12. Energy Price, O&M Costs, and Escalation Rates

Fuel

Fuel Oil: $7.50/MBtu, current cost (base)
$6.00/MBtu, 20% reduction over the base
$9.00/MBtu, 20% increase over the base

Electricity

Energy charge:	$0.0372/kWh
Demand charge:	$12/kW, Minimum charge of $12,000 for first 10,000 kW or less
Fuel adjustment Charge:	$0.05/kWh

O&M

Boilers	$1.00/1000 lb steam generated
Gas Turbines	$4.00/1000 kWh power generated
Steam turbines	$2.00/1000 kWh power generated

Escalation Rates

DOE escalation rates (for industrial region 1, discount factor = 7%)

	Mid-1981 to Mid-1985	Mid-1985 to Mid-1990	Beyond 1990	PWF 1981 to 2011
Electricity	5.27%	-1.94%	-4.07%	11.81
Fuel Oil	2.51%	2.69%	6.39%	17.79

Modified 1983 DOE Rates

	Short term 1983-1986	Long term 1986-2011	PWF 1986-2011
Electricity	0.045	-0.038	8.92
Fuel oil	0.026	0.053	20.49
O&M[a]	0.000	0.000	12.06

[a]Estimated.

Table 6-13. Energy Cost and Performance Results for Cogeneration Options with Extraction Steam Turbines for the Modified Steam Distribution System, 125 psig

Extraction Steam Turbine System[a]	Purchased Peak-Demand, kW	Purchased Electricity, $1,000	Fuel Cost, $1,000	Annual Operating Cost, $1,000	LCOC, $1,000	Energy Consumption x 1000 MBtu
T5 (3.5 MW) + 750 kW BP[b]	10,650	4,756	8,590	13,226	191,520	1,843
TR3 only	12,130	4,792	6,150	11,520	151,720	1,658
T3 + T5 (2 MW) + 3 MW BP[b]	10,068	3,740	11,259	16,010	240,070	1,468
T3 + T5 (2 MW) + 5 MW BP[b]	10,041	3,730	11,220	15,970	239,420	1,462
T3 + T5 (2 MW) + 7.5 MW BP[b]	10,014	3,720	11,190	15,930	238,910	1,457

[a] The exhaust pressure of the extration turbine is 20 in. Hga.

[b] New back-pressure steam turbine with 1235 psig.

Table 6-14. Estimated Capital Costs for Back-Pressure Steam Turbines and Gas Turbines in 1983 $

	Capacity, kW	Estimated[a] Cost, $1,000
Back-pressure steam turbines	15,000	8,750
	10,000	7,000
	7,500	5,950
	5,000	4,750
	3,000	3,750
	750	2,000
Gas turbines	18,900	11,500
	10,150	8,400
	7,400	6,250
	3,338	3,000
	2,780	2,750

[a]Total installed cost.

Table 6-15. Economics of the Cogeneration Options with Existing Turbine Combinations for the Modified Distribution System

Systems	Annual Savings[a], $1000	LCOC Savings[a], $1000	Cost $1000	SIR	Ratio of Energy saved to Capital cost, MBtu/$1000	Simple Payback, yr
T3 only	3230	73210	4900	14.9	86	1.5
T5 with 750 kW BP turbine	1520	33410	6900	4.8	43	4.5
T3 + T5 with 3 MW BP turbine	-1260	-15140	8650	b	71	b
T3 + T5 with 5 MW BP turbine	-1220	-14490	9650	b	64	b
T3 + T5 with 7.5 MW BP turbine	-1180	-13980	10850	b	58	b

[a]For the baseline case (T3 only): Annual cost = $14,750,000; life cycle cost = $224,930,000.
[b]Negative values.

Table 6-16. Results for Various Air Conditioning Combinations for the Two Turbine Cogeneration System with Fuel Price of $5.85/MBtu and DOD Escalation Rates[a]

Performance Parameter	Configuration	Case I (Ref.)[b]	Case II	Case III	Case IV	Case V
	ABS A/C, Tons	1410	0	705	2115	2820
	ELEC. A/C, Tons	3420	4830	4125	2715	2010
Annual fuel cost, $1000		4,976	4,755	4,852	5,085	5,193
Annual purchased electricity, MWh		32,370	33,690	33,030	31,720	31,060
Total annual operating cost, $1000		7,255	7,160	7,191	7,305	7,357
Annual operating cost savings, $1000[c]		0	95	64	-50	-102
Simple payback, yr[c]		N/A	14.8	11.0	N/A	N/A
LIFE CYCLE PARAMETERS:						
Life cycle operating cost (LCOC), million dollars		137.4	135.2	136.0	138.5	139.6
LCOC savings,[d] million dollars		0.0	2.2	1.4	-1.1	-2.2
Savings to investment		N/A	1.6	2.0	N/A	N/A

[a]DOD rates - escalation rate for electricity is 0.0700, and for fuel is 0.0800; discount rate is 0.10.

[b]With the existing capacities for electric (3420 tons) and absorption (1410 tons) air conditioning.

[c]Based on a replacement cost of $100/ton of air conditioning (both absorption and electric air conditioning)

[d]Compared to the reference case.

SECTION VII

FUTURE IMPROVEMENTS ON THE COGENERATION ANALYSIS PROGRAM

The CELCAP program is presently set up to analyze a cogeneration system consisting of gas turbines, diesel engines, and steam turbines. In this present form, the input information consists of hourly electric and steam loads, engine performance data, and economic parameters on fuel and electricity. The output from the program consists of the amounts of hourly electricity and steam generated and purchased, operating costs of the total system, and life-cycle operating costs of various equipment and the total system.

The following improvements are suggested for CELCAP:

(1) Life-Cycle Costs Calculations. In the present version of CELCAP only the operating costs (annual operating and life-cycle operating costs) are calculated. As an improvement to the program, the capital cost of the equipment should be included so that life-cycle costs of the cogeneration system can be calculated in the program.

(2) Heat Engine Operation. In the present version of CELCAP, in certain modes operation of the cogeneration system, the engines in the system will all be operating at part-load conditions. Improvements to CELCAP should include some logic that would make it possible for the engines to operate at full capacity before the next engine is turned on.

(3) Electric Rate Structure. The electricity rate information is provided to the program as input data in the present version of CELCAP. However, the format of this input data in CELCAP currently does not allow a complex rate structure to be included. The only way this can be included in the program is by changing several statements in the program itself. Improvements should be made in CELCAP so that more complex electric rate structures can be accommodated in input data itself.

(4) Diesel Engine Model. The model of the diesel engine included in the present version of CELCAP uses only two data points for interpolating the part-load performance. The improvements on this model should include providing more data points on the part-load performance and the use of polynomial expressions for interpolating the part-load performance.

SECTION VIII

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